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GUN EROSION.

BY LIEUTENANT COMMANDER H. E. YARNELL, U. S. N.

The problem of gun erosion is one of the most interesting and one of the most difficult that ordnance engineers have been called upon to solve. Owing to the great increase in the cost and power of modern ordnance, and the consequent increased importance of gun erosion, it has been the subject of a large amount of study and experimentation during the past few years. As in many other scientific problems, it is to a Frenchman that we owe most of our knowledge on this subject, and the treatise on erosion by M. Vieille, a translation of which follows this article, is commended to the readers of the JOURNAL as a superb example of logical and scientific analysis. This treatise was orignally published in the "Memorial des Poudres et Salpetres," 1901, 1902. The experiments described in the following article add but little to the findings of M. Vieille and confirm his results, although the method of carrying out the experiments differed to a considerable extent.

It is of interest to note that as early as 1855, Captain Noble,

F. R. S., made experiments with different metals and powders to determine their erosive capacity, and one of his results was the discovery that the milder the steel the less the erosion. The results of his and other experiments are contained in "Notes on the Construction of Ordnance, No. 46," of Dec. 26, 1887, published by the Ordnance Department of the Army.

IMPORTANCE OF THE PROBLEM.

How serious the problem is may be realized by the fact that it compelled the Ordnance Department of the Army to abandon their high-velocity 12-inch gun and to design an entirely new type of 14-inch gun with much less velocity, the loss in muzzle energy due to such decrease being compensated for by the greater weight of projectile. Such a method may be possible in ordnance mounted on shore, with a steady foundation and every facility for accurate range finding, but it is out of the question for the guns of a battleship. Such guns depend on flatness of trajectory for their ability to hit the target, and any step to reduce erosion by means of decreasing the danger space would be fatal. The strength of material that enters into the construction of modern guns is sufficient to withstand much greater pressures and velocities than are permissible for service conditions. The Brown segmental gun and the Crozier wire-wound gun, both of 6-inch caliber, were recently tested by firing 100 rounds each at an average muzzle velocity of about 3,600 foot seconds and pressures ranging as high as 30 tons per square inch. This velocity is 800 foot seconds in excess of the muzzle velocity of the present naval 6-inch gun.

The life of a present service 12-inch gun is placed at about one hundred rounds, after which it should be relined in order to maintain its accuracy. As a rate of fire of one round per minute can easily be maintained by a 12-inch gun a naval action of more than two hours' duration would seriously decrease the efficiency of the main battery. Hence, any means by which the erosion might be decreased should be given the

greatest consideration, for the advantage possessed by a fleet whose guns were in excellent condition over one whose guns were not might be vital to the fate of a nation.

CHARACTER OF EROSION.

The greatest amount of erosion exists at the origin of the rifling, and, with nitro-cellulose powders, consists of a uniform wasting away of the metal, allowing the projectile to be shoved beyond the original seat before the rotating band takes on the rifling. The deep pits and gutters produced by black and brown powders no longer take place with nitro-cellulose powders and modern gun steel. Near the middle of the bore there is no apparent erosion, and the diameter may be even less than the original diameter of the gun, which may be explained by the elongation and consequent contraction of the tube due to repeated firing. Near the muzzle the amount of erosion again increases, but it is negligible when compared with that at the origin of rifling. The wear of the bore at this place allows the shell to be shoved beyond the original seat and varies the density of loading, producing variations in the pressures and velocities at which the charges are originally fixed. As the amount of erosion will vary in the guns of the same caliber on the same ship, concentration of salvos becomes impossible with worn guns. In the case of 7-inch gun No. 1, after it had been fired 206 rounds, the erosion was so great at the origin of the rifling as to allow the shell to pass eighteen inches beyond its original seat; near the middle of the tube there was no erosion, but it began again near the muzzle, the worst part of this section being about ten feet from the muzzle. Temper cracks were present on top of the lands through the rifled bore.

RECENT EXPERIMENTS IN EROSION.

BETHLEHEM STEEL COMPANY'S EXPERIMENTS.

The following experiments were conducted by Mr. J. E. Mathews, of the Bethlehem Steel Co., at their proving grounds, Redington, Pa., March 15–30, 1905, to determine the erosive effect of nitro-cellulose gases on different metals.

A 12-inch A. P. projectile, with its base plug drilled and tapped to take the different plugs, was used. The powder was placed in the chamber of the shell, and the gases allowed to pass out through a vent in the plugs.

Composition of the Plugs.

Plug No.	Material.	C.	Mn.	Si.	Ni.	P.	S.	W.
1	Wrought iron.....	...	...	...	...	...	...	...
2	20 per cent. nickel.....	.775	.38	.186	19.57	.013	.023	...
3	Open-hearth simple steel for gun forgings.....	.48	.87	.17	...	.028	.035	...
4	Manganese-bronze.....	...	...	...	...	...	...	...
5	Nickel-steel for gun forgings.	.36	.81	.321	3.33	.026	.034	...
6	Tungsten steel.....	.29	.09	.035	...	.012	.05	3.0
7	Open-hearth for gun forgings.	Same as No. 3.						
8	Nickel-steel for gun forgings.	Same as No. 5.						
9	Tungsten steel.....	Same as No. 6.						

Three and one-fourth pounds of 3-inch, 50-caliber powder were fired in each case, and ignited in the same manner.

Results of Firings.

Plug No.	Dia. of vent before firing.	Dia. of vent after firing.	Pressure in pounds.	Appearance of plug.
1	.15625	.33	32,500	Plug worn irregularly. One very deep score.
2	.15625	.38	32,500	Plug worn irregularly, with deep pits for over half of length.
3	.15625	.33	33,500	Surface of hole smooth and polished.
4	.15625	.75	33,500	Greatly eroded. Very rough and pitted.
5	.15625	.35	31,300	Surface of hole smooth and polished.
6	.15625	.33	34,600	Surface of hole smooth and polished.
7	.15625	.34	32,000	Surface of hole smooth and polished.
8	.15625	.35	34,000	Surface of hole smooth, except one deep pit near center.
9	.15625	.34	34,000	Surface of hole smooth, except one deep pit near center.

The conclusion Mr. Mathews drew from the above results was that the open-hearth simple steel plug, the Tungsten plug and the nickel-steel gun-forging plug offered about the

same resistance to erosion, with a slight advantage in favor of the Tungsten and simple steel; that the 20 per cent. nickel-steel showed quite an appreciable increase in erosion, and that the manganese-bronze plug showed that it was entirely unsuited to be placed in contact with gases at the above temperature and pressure.

It may be noted that the conclusion drawn regarding the first three plugs is hardly borne out by the results obtained at Indian Head, where it was found that a steel with as much nickel as plug No. 5 gave quite an appreciable increase in erosion over the simple steel plug. The above method of determining the erosion by attempting to measure the diameter of the irregularly eroded hole is hardly as accurate as weighing the plugs before and after the erosion.

INDIAN HEAD EXPERIMENTS.

The following experiments were made at the Naval Proving Grounds, Indian Head, Md., March–June, 1906, to determine the effect of nitro-cellulose gases upon the amount of erosion produced in different metals, and also the effect of nitro-cellulose and other powders upon plugs of the same metal, to determine the difference of erosion caused by difference in nitration in the powders.

Method of Determining Erosion.

This was determined by weighing the plugs before and after the experiment. With metals whose specific gravities are as close together as the metals used this method gives sufficiently accurate results. With metals whose specific gravities differed by any amount it would be necessary to reduce the loss in weight to loss in volume to obtain comparative results. In arranging the plugs in order of merit for material for gun construction weight was given to the appearance of the eroded hole, whether rough and scored, or smooth and polished.

Method of Making Experiments.

A 12-inch A. P. Carpenter shell of 260 cubic inches capacity was used. The base plug was drilled and tapped to take the plugs. The same weight of powder was used in each series of experiments, and all were ignited in the same manner, 25 grams C. P. and electric igniter.

Method of Obtaining Pressures.

Three pressure gauges were used throughout the experiments. In the first experiment the gauges were placed in the bottom of the cavity underneath the powder, the shell being point downward. Very irregular pressures were obtained. The gauges were then placed on top of the powder, but it was found that one gauge would invariably be carried by the gas against the axial hole and eroded beyond use. In the succeeding experiments three holes were bored in the base plug to hold the gauges, and no further trouble was experienced.

Characteristics of Plugs Used.

The following table gives the characteristics of the plugs used:

Plug No.	Material.	Character-istics.	Dia. of hole.	Length.
1	Plain steel	Soft.	.160	4.75
2	Mild ductile steel	Soft.	.160	4.75
3	Iron	Soft.	.160	4.758
4	2 per cent. nickel-steel	Soft.	.160	4.77
5	3½ per cent. nickel-steel	Soft.	.160	4.755
6	Amorphous steel	Soft.	.160	4.758
7	Non-eroding steel	Very hard.	.160	4.792
513	Special high nickel-manganese carbon steel	Hard.	.160	4.755

The following table gives the physical characteristics of the plugs:

Plug No.	Material.	Elastic strength.	Tensile strength.	Elongation.	Contraction.
1	Plain steel.....	70,000	101,500	24.5	61.4
2	Mild ductile steel.....	36,000	66,500	34.2	64.7
3	Iron.....	39,500	54,500	26.7	39.1
4	2 per cent. nickel-steel...	64,500	97,000	24.2	54.7
5	3½ per cent. nickel-steel.	62,000	95,000	27.0	63.8
6	Amorphous steel.....	116,000	131,500	23.0	64.7
7	Non-eroding steel.....	66,000	120,000	74.3	54.4
513	Spec'l high nickel-manganese carbon steel.....	52,500*	100,000	56.5	55.7

*Apparent permanent set of .0008 at 37,500 pounds.

Chemical Analysis of the Plugs.

Plug No.	C.	Mn.	Si.	P.	S.	Ni.	Cr.
1	.471	.688	.293	.023	.026	...	...
2	.276	.756	.231	.035	.039	...	...
3	...	...	...	...	...	...	...
4	.482	.608	.259	.046	.036	2.05	...
5	.400	.661	.235	.044	.045	3.50	...
6	.293	.296	.085	.026	.024	3.59	1.89
7	.746	5.280	...	...	...	14.25	...
513	1.172	5.369	...	...	...	20.64	...

Experiments.

In the first experiment with Plug No. 1, 4 pounds of IHX-1 was used. Density of loading, .436. The fuse threads of the base plug were stripped, and the plug was blown against the roof of the explosion chamber, smashing it into several pieces. A pressure of 25 tons was given by one of the gauges. The shell showed no signs of deformation.

With the remaining plugs 1 pound of 7-inch powder (IHX-1) was used, with the following results:

Plug No.	Charge.	Pressure, in tons.	Loss of weight, in grams.
2	1 pound IHX-1	6.97	6.80
3	1 pound IHX-1	6.47	7.67
4	1 pound IHX-1	6.78	7.36
5	1 pound IHX-1	6.46	8.90
6	1 pound IHX-1	6.75	9.60
7	1 pound IHX-1	6.50	16.58
513	1 pound IHX-1	6.63	19.47

Using the same plugs, the firings were repeated with 2½ pounds of powder, with the following results:

Plug No.	Charge.	Pressure, in tons.	Loss of weight, in grams.	Remarks.
2	2½ IHX-1	18.78	24.05	Bottom of plug rounded off uniformly. Surface of hole smooth and highly polished.
3	2½ IHX-1	18.64	32.63	Bottom of plug and hole rough and scored. One deep pit near top. Hole elliptical.
4	2½ IHX-1	19.70	34.76	Bottom of plug worn irregularly. Inside of hole polished, with a few small waves near top.
5	2½ IHX-1	20.07	35.62	Same as No. 4.
6	2½ IHX-1	19.47	34.71	Bottom of plug worn off uniformly. Hole smooth and polished.
7	2½ IHX-1	19.47	53.97	Bottom of hole very irregular and pitted. Inside of hole smooth and polished, with waves from bottom to top.
513	2½ IHX-1	18.59	54.76	Bottom of hole worn uniformly. Inside of hole rough, with waves. One deep pit near top.

From the loss of weight and the appearance of the plugs, they are arranged in the following order as to their resistance to erosion:

No. 2	Best.	Mild ductile steel.
3		Amorphous steel.
4		2 per cent. nickel-steel.
5		3½ per cent. nickel-steel.
6		Iron.
7		Non-eroding steel.
513	Worst.	High-carbon steel.

These experiments proved that an increase in the percentage of carbon or nickel in steel increased the amount of erosion, and renders such alloys unsuitable for gun-tube construction.

The hardness of the metal has apparently no effect on the amount of erosion, as Plug No. 7, non-eroding steel, was almost too hard to work in the lathe, but gave nearly the greatest amount of erosion.

As shown in Vieille's experiments, the amount of erosion seems to depend on the melting point of the metal, and that it is useless to search for a remedy for erosion in some alloy. Mild steel with as small a percentage of carbon as possible

gives the greatest resistance to erosion, and is the best material for the construction of gun tubes.

EXPERIMENTS WITH POWDERS OF DIFFERENT NITRATION.

In these experiments the following experimental powders of different nitration were used:

Powder.	Grains to pound.	Nitration.
IHC—Sample A.	199	12.80
IHC—Sample B.	185	12.57
IHX—Ser. No. 10.	87	12.77
IHX—Ser. No. 11.	87	12.29
IHFF—Sample 2.	24	12.68
IHFF—Sample 1.	23½	12.34

Results of Experiments.

PLUGS OF GUN STEEL. 25 GRAMS IGNITION IN ALL FIRINGS.

Powder.	Charge.	Pressure, in tons.	Erosion, in grams.	Erosion, per lb. of powder.	Nitration.
IHC—Sam. A.....	2.50	20.80	46.22	18.49	12.80
IHC—Sam. B.....	2.50	20.45	39.78	15.91	12.57
IHC—Sam. A.....	2.253	18.76	42.13	18.70	12.80
IHX—Ser. 10.....	2.50	21.01	42.63	17.06	12.77
IHX—Ser. 11.....	2.50	19.81	36.31	14.52	12.29
IHX—Ser. 10.....	2.273	18.55	40.19	17.69	12.77
IHFF—Sam. 2...	2.50	20.69	41.42	16.57	12.68
IHFF—Sam. 1...	2.50	19.84	33.83	13.53	12.34
IHFF—Sam. 2...	2.264	18.09	36.70	16.21	12.68

PLUGS OF NICKEL GUN STEEL. SAME IGNITION.

IHC—Sam. A.....	2.50	18.79	47.09	18.33	12.80
IHC—Sam. B.....	2.50	20.36	41.05	16.42	12.57
IHC—Sam. A.....	2.253	19.12	42.37	18.81	12.80
IHX—Ser. 10.....	2.50	21.26	44.23	17.70	12.77
IHX—Ser. 11.....	2.50	18.43	35.37	14.15	12.29
IHX—Ser. 10.....	2.273	17.61	36.87	16.22	12.77
IHX—Ser. 10.....	2.50	20.86	43.77	17.51	12.77
IHX—Ser. 11.....	2.50	17.87	35.61	14.24	12.29
IHX—Ser. 10.....	2.273	18.73	40.10	17.64	12.77
IHFF—Sam. 2...	2.50	20.27	34.11	13.64	12.68
IHFF—Sam. 2...	2.50	19.65	38.42	15.35	12.68
IHFF—Sam. 1...	2.50	19.11	32.24	12.90	12.34
IHFF—Sam. 2...	2.264	17.21	34.47	15.23	12.68

The actual firings of these powders in the guns to which they belong are shown on the attached sheets.

The reduced charges of the high-nitration powders fired in the bomb were taken in the same proportion as the charges required of the high and low-nitration powders to give the same velocity in the actual gun. Thus:

2.273 lbs. IHX—Ser. 10 : 2.50 lbs. IHX—Ser. 11 = 60 : 66.

This was hardly necessary, as the result showed that the erosion per pound of the same powder is the same after the pressure has attained sufficiently great limits.

From the above results it is seen that—

(1) The erosion per pound of any one powder remains practically the same under sufficiently great pressures.

(2) That by decreasing the amount of nitration the amount of erosion is decreased.

(3) That in spite of the smaller charge necessary for a high-nitration powder, the actual amount of erosion is still greater.

(4) That the decrease in erosion does not follow the same ratio as the decrease in heat units in differently nitrated powders. This was proved by obtaining the heat units of the 7-inch powder in a calorimeter. While the heat units in the smaller charges of the higher nitration powder were less than the total amount of heat units in the charge of lower nitration, the erosion was found by experiment to be greater.

The above results lead to the question whether the increased ballistic efficiency of the higher nitration powder counterbalances the increased amount of erosion.

A comparison of the charges necessary to give the service velocity of the above powders is as follows:

Powder.	Velocity.	Charge.	Nitration.
IHC—Sample A.	2,150	20.3	12.80=Dif. .23
IHC—Sample B.	2,150 c.	22.5=2.2	12.57
IHX—Ser. 10.	2,700	60.0	12.77=Dif. .48
IHX—Ser. 10.	2,700	66.0=6.0	12.29
IHFF—Sample 2.	2,600	298.0	12.68=Dif. .34
IHFF—Sample 1.	2,600	329.0=3.1	12.34

From the above it is seen that the gain in ballistic efficiency by increasing the nitration of the powder is more apparent than real, since the above powders have wide limits of nitration.

In view of these experiments, it would seem that more is gained in reducing the amount of erosion by lowering the nitration than is lost in the decreased ballistic efficiency of the powder.

SUMMARY.

The following gives the conclusions that may be drawn from the above experiments:

Metals.

(1) The present mild gun steel gives the least erosion of any of the iron alloys that are suitable for gun construction.

(2) Increasing the amount of carbon in the steel greatly increases the amount of erosion.

(3) Alloying steel with manganese, chromium, etc., increases the amount of erosion.

(4) Any special treatment for the purpose of giving hardness to the steel has no effect in reducing erosion.

Powders.

(1) Increasing the nitration of the powder increases its erosive power.

(2) This increased amount of erosion is greater than can be compensated for by the decreased charge of the higher nitration powder.

OTHER EXPERIMENTS.

The following experiments were made with the object of determining whether the amount of erosion could be decreased by the addition of any substance to the powder charge, and also to ascertain the effect of plugging the axial hole, thus making the gases do work before escaping.

PLUGS OF GAS-CHECK PAD METAL. HOLES 0.152 INCH IN DIAMETER. LENGTH 5.25 INCHES.

Plug.	Powder.	Charge.	Pressure, in tons.	Erosion, in grams.	Erosion, per lb. of powder.	Remarks.
KK	CSPP-8	1.75	14.64	33.42	19.10	Hole plugged.
LL	CSPP-8	1.75	14.17	34.02	19.44	Hole open.
MM	CSPP-8	1.75	14.19	33.14	18.94	Hole open.
OO	CSPP-8	1.75	15.33	9.32	5.32	6 ounces water.
GG	CSPP-8	1.75	15.19	10.13	5.79	6 ounces water.
FF	CSPP-8	1.75	15.13	27.34	15.62	6 ounces water.
II	CSPP-8	1.75	15.27	22.62	12.92	3 ounces paraffin.
JJ	CSPP-8	1.75	15.05	20.51	11.72	1.5 ounces paraffin.
HH	CSPP-8	1.75	15.40	34.49	19.71	Water in bottom.

Plug KK.—The hole was stopped with a steel wire, which extended through into the cavity of a 3-pounder shell outside the bomb. The wire was supported by a piece of hard wood driven into the cavity of the shell and having an axial hole just large enough to take the wire.

Plugs LL and MM.—These firings were made for comparison with KK, the axial hole being open. From the resulting erosion, it is seen that the amount with the hole plugged up does not differ materially from that obtained with the hole open, one of the latter results being greater and the other smaller than with plug KK.

Plugs OO, GG and FF.—In these firings a can containing about six ounces of water was placed on top of the powder. The can was closed, except two small openings in the top. In plugs OO and GG the can had evidently been carried against the axial hole with the first burst of flame and held there during the escape of the gas, as both tins were found welded to the head of the erosion plugs after the firing. In FF the can was found almost intact in the bomb after firing, although the water was evaporated.

In the cases of OO and GG a very efficient distribution of the water had taken place, as shown by the low amount of erosion and the welding of the cans to the erosion plug.

With FF an entirely different action evidently took place. The water remained behind the greater mass of the gas, and, although evaporated, could do but little towards cooling the walls of the axial hole.

Plug II.—A solid piece of paraffin weighing about three ounces was placed on top of the powder. $1\frac{3}{4}$ ounces were recovered unmelted after the firing.

Plug JJ.—$1\frac{1}{2}$ ounces of powdered paraffin were placed on top of the powder.

Plug HH.—Six ounces of water were placed in the bottom of the bomb under the powder charge. Very little of the water was evaporated, and there was no reduction in the amount of erosion.

The pressures were invariably higher in all the firings with water and paraffin, showing that the cooling effect of these substances did not affect in any manner the pressure of the resulting gases.

The increased pressure obtained was probably due to the decreased air space of the chamber by the additional volume of water, and consequent increased density of loading.

CAUSES OF EROSION.

The conclusion drawn by M. Vieille, which was fully confirmed by the preceding experiments, was that gun erosion is caused by the escape of the highly heated gases of combustion through the minute openings between the rotating bands and the bore of the gun before and shortly after the projectile begins to move. After the shell has advanced sufficiently far to enable the copper rotating band to mold itself to the form of the rifling the erosion decreases to a considerable extent. The pressure that is reached in the powder chamber before the projectile begins to move is not definitely known, but may be assumed at about four tons per square inch, which produces an enormous velocity of flow of heated gas past the projectile.

The amount of erosion, as shown by M. Vieille, depends upon—

(1) The melting point of the metal;

(2) The temperature of combustion of the powder; or,

The least amount of erosion is obtained with the gun metal having the highest melting point and the explosive having the lowest temperature of combustion.

In this connection the carbonizing effect of the carbon monoxide of nitro-cellulose gases under pressure upon the gun metal may increase the susceptibility of the metal to erosion, since it has been shown in the preceding experiments that an increase in the amount of carbon in the steel increases the erosion. This point was first brought out by Mr. G. W. Patterson, Chief Chemist of the U. S. Naval Proving Ground, in the following memorandum:

In 1892 some experiments were performed at the Torpedo Station in regard to the use of carbon monoxide as an agent for carbonizing armor plates. Billets of mild machine steel containing about 0.15 per cent. carbon were enclosed in a tight iron chamber and heated in a furnace in an atmosphere of carbon monoxide under pressure, up to 165 pounds per square inch.

From the results of these tests it was shown that the steel was capable of carbonization and also of decarbonization. In some cases carbon content was increased 100 per cent. and in others decreased by 50 per cent.; but the heat conditions determining these effects were not worked out. In billets after treatment the color of the steel was a silver-white and the surface presented a fretted appearance.

In Howe's "Metallurgy of Steel," Vol. I, p. 120, of 1890, is cited experiments showing the carbonization of metallic iron, and the statement is made that the effect is produced most rapidly at 400–450 degrees C., and ceases at or above bright redness; that the effect may be uniform or take place in patches.

I consider that the erosion is best accounted for by a rapid carbonization and decarbonization of the surface of the steel by the carbon monoxide of the powder gases, with constant roughening of the surface, as it is then in the most favorable

condition for being worn away by the blast of the intensely-heated gas.

Mr. Patterson cited in proof of his theory that in the case of fragments of guns that had burst after repeated firing the metal next the bore was exceedingly hard and could barely be cut with a file, showing that its composition had been changed by the action of the powder gases. The metal which has undergone this change of composition may be from $\frac{1}{16}$ to $\frac{1}{8}$ inch in thickness, as found from the fragments, and presents a dark appearance in contrast to the silver gray of the original metal.

CAN EROSION BE REDUCED?

From an examination of the causes of erosion and the preceding experiments, any investigation leading to a reduction of erosion would necessarily be along the following lines:

(1) To obtain a gun metal having the highest possible melting point.

(2) To obtain an explosive having the lowest possible temperature of combustion.

(3) Mechanical means for reducing the erosion.

With regard to the suitable gun metal, it would seem that the present mild gun steel, with the smallest percentage of carbon possible, is the best that can be obtained. Any alloy, as nickel, manganese or tungsten, lowers the melting point and increases the susceptibility of the tube to erosion.

The problem of obtaining a powder which will give the required propellant effect and still have a low temperature of combustion offers a great field for investigation. The experiments of M. Vieille with nitroguanidine are most interesting, but no information is given as to the efficiency of the explosive with which it was combined. It seems to be a general rule that the temperature of combustion increases in reverse ratio to the weight of the charge. The propulsive efficiency of cordite, with its low weight of charge and consequent smaller volume of powder chamber and increased length of rifled bore, is more than counterbalanced by the high temper-

ature of combustion and erosive effect. In our own nitrocellulose powders a smaller charge may be obtained by increasing the nitration, but such increase means higher temperature of combustion and increased amount of erosion.

With regard to the third method of prevening or reducing erosion by mechanical means, all attempts naturally fall under the method of obtaining a more effective seal to the escape of gas before and shortly after the projectile starts to move. The preceding experiments, in which the amount of erosion was reduced over 300 per cent. by the simple means of introducing water into the powder chamber, might lead to valuable results in actual firing. Any circular receptacle which would contain the fluid and which would burst upon the shock of discharge would serve the purpose. As the necessity for such a seal exists for an exceedingly small fraction of a second, not much liquid would be required, and its high latent heat would effectively cool such small percentage of the total volume of gas as would escape past the projectile. The receptacle could be designed to act as a seal in itself. In this connection it is interesting to note that a 6-inch, 40-caliber gun at the proving ground showed no appreciable erosion after having been fired 1,700 rounds. In this type of gun the charge is contained in a metallic case which is sealed with a copper "mouth cup." It was this "mouth cup," in addition to the regular rotating band, which contributed to this remarkable record.

By using paraffin in one of the experiments it was found that the erosion was reduced nearly 100 per cent. In this connection it was found that the guns of one of the ships were eroded considerably less than those of a sister ship after an equal number of rounds, due to the fact that on the former a considerable amount of vaseline was used on the projectile.

Another method of reducing erosion is to use as slow a powder as possible, thus moving the point of maximum pressure farther along the bore and decreasing the amount of chamber pressure when the projectile starts to move. The limit of slowness of the powder is reached when complete combustion does not take place in the bore of the gun.

A STUDY OF EROSION.

By M. VIEILLE.

Translated by LIEUT. H. E. YARNELL, U. S. N., from "Memorial des Poudres et Salpetres."

The use of certain colloidal powders for the past fifteen years as a propellant in cannon has resulted in damage to the bore by erosion, which quickly destroys the accuracy of fire and renders the gun useless after a very limited number of rounds.

This fault occurs especially with powders having a nitroglycerine base, and it is on this account that powders of this type have not been adopted by most European countries. Yet these powders have such remarkable ballistic properties that if it were possible to avoid erosion by a suitable choice of gun metal, or by a modification of the composition of the powder without altering its ballistic properties, a great step in advance would immediately be attained.

The direct experiments on metals or powders by actual firing in a gun is impracticable, not only on account of the expense involved, but because such a method would not be suitable for a methodical study of the very numerous alloys which now can only be furnished in small quantities; besides, such a method would not lend itself to a systematic study of types of new powders, which are easy to make in small quantities in a laboratory, but of which the commercial manufacture would not be undertaken without preliminary tests having shown their advantage.

The attempt by means of some simple means of experiment to classify the different metals and alloys according to their susceptibility to erosion, and the different powders ac-

cording to their erosive power, presents, therefore, a subject of undoubted interest.

Experiments of such a nature have been carried out in different countries, especially in England by Armstrong, and the superiority, from the erosive point of view, of mild steel over other metals and alloys has particularly been established. Nevertheless, it has appeared useful to coördinate these results by methodical experiments; that is the object of the present treatise.

CHARACTER OF BALLISTIC EROSION.

It will be best, first of all, to explain the nature of the phenomenon which it will be attempted to reproduce.

The erosion produced in the bore of guns may be examined either by cutting the gun in two longitudinally, or by means of gutta percha or plaster casts. The erosion is usually greatest at the origin of the rifling, and extends to that part of the bore passed over by the shell at a slow velocity and under the highest pressures.

The beginning of the erosion appears to be a network of fine cracks which cover the chamber in the neighborhood of the compression slope. The depth and width of the longitudinal cracks increase with the number of shots fired, and their junction forms a new surface of the bore, on which stands out a network of islets whose tops appear to mark the original surface that has been eroded.

The following photographs show very clearly the transverse and longitudinal system of cracks in the chamber and the predominance of longitudinal striae parallel to the axis of the gun beyond the origin of the rifling.

The origin of the network of original cracks may perhaps be explained in different ways. It seems to me most probable that it is caused by the superficial carburization and high temper assumed by the surface of mild steel when subjected to the action of carbonic gases such as arise from the decomposition of explosives. This carburization and temper is con-

stantly observed in the metal of closed chambers. This thin film of tempered and carburized metal easily submits to this process of cracking under the action of the high pressure and temperature which it undergoes in a gun, and this first phase of erosion, resulting in a quadrilateral series of cracks, is entirely independent of the leakage of gas, which only acts to increase the striae parallel to the bore of the gun.

The leakage of gas is due to the fact that the rotating band is not able to mold itself in the fine crevices. The gas therefore tends to escape from the powder chamber through these passages past the projectile into the bore with a velocity far greater than that of the projectile itself. These cracks parallel to the axis of the bore form the most direct line of escape for the gases, and they are in fact the ones which are found to undergo the most rapid erosion.

It must be borne in mind that the duration of this leakage of gas is very short and only exists at any one place during the passage of the rotating band. It is therefore bound to be greater where the velocity of the shell is small and the pressure great, and for that reason the erosion is found limited within a space from one to two calibers' distance from the origin of the rifling. Beyond that the velocity of the projectile is too great and the pressure too much reduced to allow any appreciable erosion to result from the leakage through very small orifices of gas even of very high temperature.

Whatever the value of the above explanation it is certain that erosion cannot be attributed to the effect of whirlwinds of gas during the burning of the charge, since it is not observed either in a closed chamber or in that part of the powder chamber near the breech plug.

Nor can it be attributed to the friction of the gas against the walls of the gun, even with velocities far greater than that of the projectile itself, since there is no erosion observed just inside the muzzle, where great leakage again takes place on account of the lack of obturation just as the projectile leaves the gun.

The description which has just been given is that of the

ordinary and prevalent form of erosion. This ordinary form may sometimes be varied by the formation of very large pits in the surface of the bore, arising, probably, from lack of homogeneity in the gun metal or in some local fault in the obturation of the rotating band. These irregularly-disposed spots may be eroded by continued firing to a depth of several millimeters.

Experience has shown that these abnormal and unusual forms of erosion may be avoided by a suitable design of rotating band, whereas the ordinary form of erosion defined above will remain as an unavoidable result of prolonged fire until some new metal or propellant is discovered.

In brief, it is the repeated action of a jet of gas of high temperature escaping freely past the projectile through a very small orifice to which the phenomenon of erosion may be attributed.

The first chapter of the work is devoted to a description of the method of experimentation adopted and to a study of the variable that would affect the value of the results. Through this study has been determined the most favorable conditions for the accuracy of the measurements, and thereby to determine the practical results of the experiments.

The second chapter shows the susceptibility to erosion of a large number of metals and alloys.

The conclusion to be drawn from this long series of experiments is that there is no relation between the physical properties of the metals, their hardness, their elastic limit and their susceptibility to erosion. This latter property only appears to depend on the melting point, at least for the iron group of metals. Pure iron and steel with a very small percentage of carbon, offer the greatest resistance to erosion, and, under this condition of a high melting point, all these steels are equal.

Consequently, it seems useless to hope for a solution of the problem of erosion caused by powders having a nitroglycerine base, by means of some special metal.

In a third chapter I have studied the different types of

powders and explosives. Their erosive powers vary within very wide limits, the amount of erosion increasing ten-fold in passing from nitroguanidine to ballistite, not to speak only of explosives whose forces are about of the same order, other things being equal.

It is, therefore, in the choice of the explosive and in the rational combination of material of different erosive powers that we must search for the remedy for the destructive effect of certain explosives having a high ballistic value.

In a fourth chapter we have combined the data, showing the physical nature of erosion caused by an escaping gas of high temperature and great velocity. This résumé contains a theory of the phenomenon, which may be, however, but a mere approximation.

In a note which ends this chapter M. Osmond has very kindly contributed some interesting observations on the molecular changes which accompany the superficial tempering of eroded steel.

CHAPTER I.

CHOICE OF AN APPARATUS FOR MAKING THE TESTS.

After what has been said above regarding the nature of erosion in guns, in order to simulate practical conditions the experimental apparatus should be operated with a great leakage of gas through a small orifice.

It is equally necessary, in order that the device may be a true experimental apparatus, that the experiments may be repeated and extended without difficulty to explosives and metals prepared only in the form of samples and in consequence of which the weight of the metal specimen eroded is small in comparison with the charge of explosive necessary to produce an erosion that can be measured with accuracy.

These conditions were realized through the use of metal cylinders about 22 millimeters in diameter, with an axial hole 1 millimeter in diameter through which the gas escaped. The length of the hole and specimen has usually been about

40 millimeters. The drilling of such small holes, a very difficult matter, was done at the Laboratoire Central de la Marine, to which we are indebted for the most generous assistance in the prosecution of these experiments.

Specimens 20 millimeters and even 10 millimeters in length are easier to prepare and, as soon will be seen, are capable of classifying the explosives in the same manner; but the absolute value of the erosion is reduced in large proportions, and the accuracy of measurement is found to be diminished.

METHOD OF MAKING THE EXPERIMENTS.

The greater part of the experiments during this research have been made by employing a steel chamber with three necks, represented in the following sketch (See Fig. 3).

This chamber is closed by three steel plugs. Plug A serves as a firing lock. It has a cylindrical conical hole, large end inside, in which is very accurately fitted a piston which forms a stop valve, and is insulated with a coat of shellac. This piston forms the leading wire of the firing battery, whose current passes through and incandesces a fine iron wire in an igniter of very fine black sporting powder. The other end of the igniting wire is connected to plug A. The joint between plug A and the bomb is sealed by a copper gas-check ring *bb*. The head of the crusher-gauge piston, or the flexible bar of the spring gauge is fitted with a steel pen or tracer which inscribes the movement of the piston upon a revolving cylinder. Thus we obtain the rate of increase of pressure inside the bomb as a function of the time.

Plug C, which holds the test specimen, contains an axial hole about ten millimeters in diameter, which permits the gas to escape freely after having passed through the test specimen.

The volume of this bomb is about 17.8 cubic centimeters. The weight of powder which may be burned varies from 3.5 grams to 10 grams. I have also used cylindrical bombs of much greater capacity—75 and 340 cubic centimeters (see Figs. 4 and 5). These bombs have but two necks, one serv-

ing as the firing-plug end and the other as the receptacle for the test specimen. The charges which can be burned in these bombs are five and twenty times greater, respectively, than can be burned in the small one.

Fig. 3.

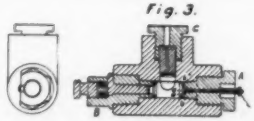


Fig. 4.

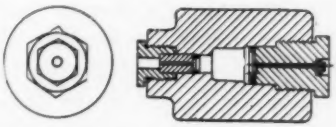
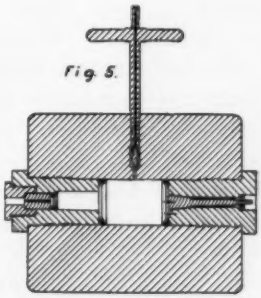
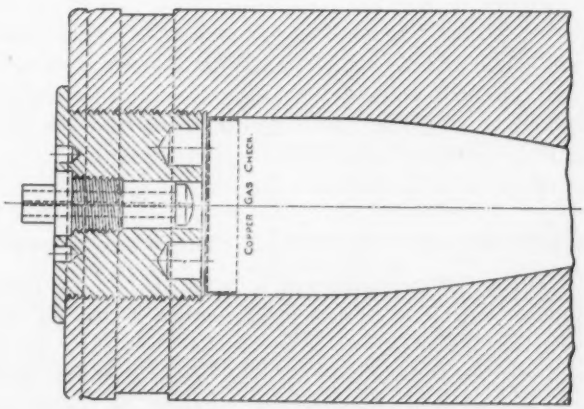


Fig. 5.





Under these conditions of experimentation the burning of the charge gives an explosion that may be compared to the heavy discharge of a gun, and the experiments may be dema with the three types of bomb in a closed shooting gallery without inconvenience to the neighborhood. This is on account of the small diameter of the axial hole of the test spec-

imen. It was discovered accidentally in an experiment made with a test specimen having an axial hole 10 millimeters in diameter, mounted in the 340 cubic-centimeter bomb, that the results were rather violent, breaking many of the windows of the gallery and causing an almost negligible amount of erosion.

MEASURING THE EROSION.

The test specimens through which the gas escapes weighed about 135 grams. It is weighed within $\frac{1}{1000}$ of a gram before the experiment. After the experiment it is thoroughly wiped off and weighed again with the same care. The difference in weight varied in the different experiments from .15 gram to 8 or 9 grams. In every case it was obtained within $\frac{1}{100}$ of a gram. This variation in weight may be transformed into volume by dividing the weight by the density of the metal, say 7.8 in the case of ordinary steels. This transformation is useful in comparing the action of metals of different densities. In some experiments the transformation has been made directly with the use of mercury, subtracting from the observed volume the original volume of the axial hole, which is about 36 cubic centimeters. The results obtained by this more difficult method are identical with the results of the calculations made after the weighings.

MEASURING THE MAXIMUM PRESSURE AND THE RATE OF ITS INCREASE.

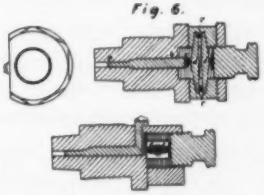
The crusher gauge consists of a cylindrical piston from .5 to 1 square centimeter area of cross section, according to the pressure, the base of which receives the pressure of the gas. Its head compresses a copper cylinder about 8 millimeters in diameter and about 13 millimeters in height, of the standard navy type. The compression and consequent pressure are picked off from the tarage table prepared at Gavres.

A pen attached to the head of the piston inscribes the pressure curve upon a revolving cylinder.

The spring gauge is composed of a piston *ab*, the small or pressure end having an area of about 14 square millimeters. The other end is enlarged and ends in a hemispherical cap

which abuts against the spring arrangement. The latter consists of two bars shaped to form a trapezoid, which make them almost equal to a solid in resistance. These bars are yoked together by means of two small steel pins about 2 millimeters in diameter, placed in shallow, transverse grooves at

Fig. 6.



the ends of the bars. These bars, indicated by 5/2–5–40, have a length of 40 millimeters between the grooves, a width and thickness of 5 millimeters at the center and a thickness of 2 millimeters at the ends.

The tarage curve of this spring is obtained in the same manner as with a spiral one, by means of the free-piston tarage machine.

These springs, which are very light and easy to make, give without breaking a compression of 1.9 millimeters when they are in contact at the middle of the bars.

The compression is proportional to the pressure. The pair which I used in these experiments gave a compression of 1 millimeter for 270.2 kilograms' pressure.

The small piston *ab* works without any metallic gas check, a plug of tallow about 3 millimeters thick being used instead.

By means of the spring gauge can be obtained the curve of increasing and of decreasing pressures. Its very small period of oscillation allows it to follow statically the very rapid variations in pressure.

METHOD FOLLOWED IN THE EXPERIMENTS.

In order to be able to understand thoroughly the results of the experiments made under the foregoing conditions it is well to study the effects on the results of variations in the following elements:

1. Amount of charge.
2. Repetition of firings with the same specimen.
3. Maximum pressure and variations in combustion of the explosive.
4. Diameter of axial hole of specimen.
5. Length of axial hole.

The following experiments were made to determine the variations resulting from changes in the above:

I. Effect of Changes in Diameter of Axial Hole.

Comparative experiments were made on specimens of gun steel 40 millimeters long, the axial holes having diameters of 4 millimeters, 2 millimeters and 1 millimeter.

The erosion was produced by the explosion of 3.55 grams of a quick, small-arm powder, BF–AM (pure nitrocellulose), and with VF, a powder of the ballistite type containing 50 per cent. of nitroglycerine.

These powders were exploded in the 17.8 cubic-centimeter bomb; density of loading, .20; maximum pressure, about 2,500 kilograms.

The resulting erosion, found by loss of weight and reducing this loss to cubic millimeters, is as follows:

Kind of powder.	Hole, 4 mm. dia.		Hole, 2 mm. dia.		Hole, 1 mm. dia.	
	Eros.	Press.	Eros.	Press.	Eros.	Press.
BF, 3e, 91, SL or BF, AM46e, 98R. Colloid of pure nitrocellulose.	.20	1,244	6.9	2,039	20.3 22.7	2,403 2,361
	.50	916	8.5	1,938	24.7 25.5	2,224 2,253
VF—Powder of ballistite type. 50 per cent. nitroglycerine.	12.70	1,084	58.6	2,186	84.5 83.2	2,509 2,370
	12.30	1,268	58.2	2,099	90.2 85.9	2,542 2,360

Erosions in cubic millimeters.
Pressures in kilograms per square centimeter.

It appears from the above results that the same charge of powder produces a rapidly increasing amount of erosion as the axial hole becomes smaller. The reduction of erosion

with holes of larger diameter is not wholly caused by the reduction in maximum pressure, since it is found that while in the 2-millimeter and 1-millimeter specimens the amount of erosion differed greatly, the difference in pressure was very small, and besides, from direct experiments it has been found that when the pressure reaches 2,000 kilograms (about 5 tons) the increase in erosion through increase of pressure is almost negligible.

The reduction in erosion results principally from the fact that the gas only erodes around the periphery of the escaping jet, and as the volumes of gas given off by the same charges are approximately constant, the active surfaces of the jets are inversely proportional to the diameters of the axial holes.

The amount of erosion, therefore, becomes much greater as the hole becomes smaller, and it is of value to determine the minimum orifice. It would seem that the minimum practical diameter is fixed at about 1 millimeter, as the drilling of a smaller hole for a considerable length is a matter of great difficulty.

II. Effect of Length of Axial Hole.

On account of the difficulty of drilling holes 1 millimeter in diameter through specimens of considerable length experiments were undertaken to determine the effect of the length on the amount of metal eroded.

The following experiments were made with specimens the axial holes of which were 1 millimeter in diameter, and the lengths 10 millimeters, 20 millimeters, 40 millimeters, respectively.

The powders used were the same as used in the preceding experiment—pure nitrocellulose powder and ballistite. Charges were from 3.55 to 3.35 grams. Same method of ignition. Maximum pressure about 2,500 kilograms (about 6.2 tons).

Kind of powder and charge.	Specimens 10 mm. long.		Specimens 20 mm. long.		Specimens 40 mm. long.	
	Eros.	Press.	Eros.	Press.	Eros.	Press.
BF–AM, pure nitrocellulose :						
Charge, 3.35 + .60 grams...	9.2	2,200	.....	.....	19 5	2,162
Fine-grain ignition :						
Charge, 3.55 + .10 grams...	11.4	2,343	13.8	2,282	24.7	2,224
3.55 + .10 grams...	11.0	2,234	.....	.....	.....	.....
VF–Ballistite, 50 per cent. nitroglycerine :						
Charge, 3.55 + .10 grams...	39.9	2,380	57.3	2,485	90.2	2,542
3.55 + .10 grams...	33.8	2,476	.....	.....	90.2	2,542
CNF*, cordite combined with nitroguanidine :					30.2	2,375
Charge, 3.55 + .10 grams...	11.92	2,504	.....	.....	26.1	2,244
Small-arm cordite, Sevran, 1894 :						
Charge, 3.55 + .10 grams...	26.6	2,607	.....	.....	64 2	2,500

*In the CNF powder the temperature of combustion was reduced by the addition of nitroguanidine.

The ignition in the above firings was all fine grain.

The above results show the desirability of increasing the length of the axial hole as much as possible, having regard for the difficulty of drilling, since the amount of metal eroded for the same charge increases in the ratio of 1 to 2.5 in passing from a specimen 10 millimeters long to one 40 millimeters long.

These results also call for a second remark, which is that the classification of metals according to their erosive power remains the same regardless of the length of the axial hole, even though this erosive power varies in the ratio of 1 to 4 in passing from pure nitrocellulose powders to ballistite with its 50 per cent. of nitroglycerine.

III. Effect of the Quantity of Gas Passing through the Axial Hole under the Same Maximum Pressure.

This effect may be studied by using similar test specimens in bombs of different sizes, the charges being regulated to give the same density of loading and consequently the same

maximum pressure. Bombs of 343.7 cubic centimeters, 73.8 cubic centimeters and 17.8 cubic centimeters were used.

The respective volumes and charges varied in the ratio of 1 to 20. The maximum pressure was from 2,200 to 2,500 kilograms.

The following comparative results were obtained with specimens of gun steel.

Kind of powder.	△	343.7 bomb.		73.8 bomb.		17.8 bomb.	
		Erosion.	Eros. per gram.	Erosion.	Eros. per gram.	Erosion.	Eros. per gram.
BF.....	.20	393 392 374	5.5	109.2 105.1	7.3	20.1 20.3 22.7 24.7 25.5	6.4
English small-arm cordite.....	.20	738	10.2	...	...	64.2	18.0
VF.	.20	1,014	14.7	322.2	21.5	83.5 83.2	24.9

NOTE.—Amount of erosion expressed in cubic millimeters.

From the above it is seen that the amount of metal eroded per gram of explosive becomes greater as the charge becomes smaller, although the actual amount of metal eroded becomes smaller; and that the ratios of their erosive powers, on the basis of the amount of erosion produced per gram of explosive, are the same in the three series of experiments.

The small bomb of 17.8 cubic-centimeter capacity shows the most clearly the difference in action of the three explosives. Indeed, this result could have been foretold. The decrease in the amount of erosion per gram of explosive in using larger charges and bombs is the evident result of the increase in diameter of the axial hole, the results of such variations being shown in Par. 1. Where the erosion is very great a large part of the stream of gas produces no effect.

IV. Effect of Repetitions of Firings with the Same Specimen.

We have seen that our theory as to the process of erosion in guns requires the repetition of escaping jets of gas through orifices of small section.

Consequently, it is necessary to show how the classification of metals and explosives determined by the erosion of a single firing would be modified by repeated firings.

These experiments have been made with the 17.8 cubic-centimeter bomb and results obtained of the amount of erosion caused by repeated firings with the same powder charge, using the same test specimen in each case, pressures being taken each time with the pressure gauge.

The experiments were carried out with the same two powders as were used in the preceding series—BF–AM and VF. Experiments were also made with a black sporting powder, an Italian small-arm powder with a nitroglycerine base, called solenite, and finally with a dynamite, 75 per cent. nitroglycerine and 25 per cent. kieselguhr.

The table on opposite page gives the results obtained. The test specimens were of gun steel, the amount of erosion being given by weight in milligrams.

In these tables the observed erosions have been compared with calculated results by assuming that the erosion varies inversely as the number of firings,

$$E = \frac{e}{\sqrt{N}}$$

The results are in close accord for the BF–AM powder, the solenite, the dynamite and even the black sporting powder, in spite of the erratic result of the third experiment.

With ballistite the above law of decrease is not sufficiently great. But on the whole, in spite of a few anomalies, these results show that the consequent erosion produced by repeated firings follow almost exactly the same law for all powders, and that the classification of metals and powders from single firings is as accurate as would be obtained by re-

peated firings with each of the types of metals or powder. It is also seen that the observed maximum pressures decrease gradually for all the powders with the exception of the black. This shows the increasing effect of the jet of gas which escapes during the time required to reach the maximum pressure.

No. of firing and powder.	Erosion, in milligrams.		Pressure, in kilograms per square centimeter.
	Observed.	Calculated.	
1. Powder, BF–AM. △ = .20..	193	193	2,224
2. " "	134	136	2,215
3. " "	103	111	2,014
4. " "	98	96.5	2,014
5. " "	103	86.3	2,044
6. " "	71	78.8	2,084
7. " "	62	72.9	1,994
8. " "	65	68	1,742
9. " "	56	64	1,860
10. " "	49	61	1,855
1. Powder, VF (Ballistite). △ = .20..	704	704	2,542
2. " "	549	498	2,084
3. " "	477	406	1,989
4. " "	423	352	1,725
5. " "	338	315	1,628
1. Solenite. △ = .20.....	381	381	2,449
2. " "	266	269	2,286
3. " "	212	220	2,214
4. " "	174	190	2,138
5. " "	172	170	2,114
1. Black sporting powder. △ = .50.	312	312	1,839
2. " "	203	220	1,958
3. " "	99	180	1,860
4. " "	132	156	1,881
1. Dynamite. △ = .20.....	470	470	2,084
2. " "	324	332	1,802
3. " "	249	271	1,785
4. " "	227	235	1,855

For the black sporting powder, which was the quickest and most inflammable of those used in the experiment, this initial escape of gas before the attainment of maximum pressure is even until the fourth firing, thus resulting in the pressures remaining about equal.

From the above it may be stated that the gradual reduction in erosion results from two causes—one on account of the increase in diameter of the axial hole, decreasing the amount of metal eroded, as has been shown in direct experiments; the other on account of the decreasing pressure under which the gases first escape, the initial time of the escape of gas, when its effect from an erosive point of view is small, becomes predominant.

In brief, it may be assumed that the effects of continued firing such as result in the actual gun follow the same law as the results that can be obtained from a single experiment.

V. Effect of the Pressure.

In all the preceding experiments we have studied the effect on erosion of variations in the length and diameter of the axial hole, where the gases reached practically the same maximum pressure, and where, on account of the quickness of the powder, the maximum pressure was reached before the amount of escaping gas attained proportions that need be considered. These maximum pressures were from 2,000 to 2,500 kilograms per square centimeter.

The question now is to determine the actual value of the effect of this pressure. This effect will be found combined with certain complex actions which are rather difficult to analyze.

Effect of the Pressure, the Charge Remaining Constant.

One method of varying the maximum pressures without varying the charge is to alter the capacity of the powder chamber of the bomb, thus varying the density of loading. The capacities of chambers used were as follows: 11.8 cubic centimeters, 17.8 cubic centimeters, 49.7 cubic centimeters, and 343 cubic centimeters, thus in ratios from 1 to about 30 in capacity.

The capacity of 11.8 cubic centimeters was obtained by reducing the capacity of the 17.8 cubic-centimeter bomb with

a large-sized copper gas check, and also with a known weight of molten sulphur which was poured in the bottom of the bomb and covered with a layer of putty, this all being found intact after the explosion.

The chamber of the 49.7 cubic-centimeter bomb was obtained from the 17.8 cubic-centimeter bomb by the use of a shorter firing plug. The volume of 343 cubic centimeters was that of the large bomb.

The results obtained are given in the following table. The charges are for a density of loading of .20 in the 17.8 bomb.

Kind of powder.	343 cu.-cm. bomb.			49.7 cu.-cm. bomb.		
	Charge.	Eros.	Press.	Charge.	Eros.	Press.
BF or BF-AM.....	3.55	.30	100	3.90	5.9	906
VF (ballistite).....	3.55	.00	100	3.50	40.6	879

Kind of powder.	17.8 cu.-cm. bomb.			11.8 cu.-cm. bomb.		
	Charge.	Eros.	Press.	Charge.	Eros.	Press.
BF or BF-AM.....	3.45	20.3	2,403			
	3.45	22.7	2,361			
	3.55	24.7	2,224	3.55	30.6	4,128
	3.55	25.5	2,253	3.55	26.4	3,876
	3.55	20.1	2,143			
VF (ballistite).....	3.45	84.5	2,509	3.55	85.7	3,807
	3.45	83.2	2,370			
CNF (Eng. cordite).....	3.55	64.2	2,500	3.55	64.5	3,710

Charges are in grams; erosions in cubic millimeters; pressures in kilograms per square centimeter.

The above experiments show that the erosion caused by the same mass of gas is negligible with pressures under 100 kilograms; that it increases in a considerable ratio when the pressure is raised from 900 to 2,000 or 2,500 kilograms, but that the amount of erosion remains almost stationary in raising the pressure from 2,000 to 4,000 kilograms.

Effect of the Pressure with Increasing Charge in a Constant Volume.

The pressure may be obtained by keeping the volume of the powder chamber constant and increasing the charge.

In this case the amount of erosion varies both with the charge and the pressure; however, it has been previously determined that the amount of eroded metal per gram of powder burning under the same maximum pressure was very slightly affected by varying the charge within wide limits, say in the ratio of 1 to 5.

In a chamber of the same capacity a much smaller ratio than 1 to 5 is needed, as an increase of charge in the proportion of 2 to 3 will double the pressure.

Consequently, if under these conditions the erosive power of the explosive remains constant the pressure has no effect upon the erosion within the limits assumed for comparison.

Following this method comparisons have been made of the erosion produced by charges of 3.55 and 5 grams in the 17.8 cubic-centimeter bomb, giving densities of loading of approximately .20 and .30.

The following table gives the results of these experiments:

Kind of powder.	△	Pressure.	Erosion in mm³.	Erosive power per gram of powder.
BF–AM	.20	2,253	25.5	7.20
	.281	3,500	41.1	8.23
	.281	3,614	40.0	8.00
VF	.20	2,360	85.9	24.2
	.281	3,886	124.8	24.4

Although the pressures have been nearly doubled, and the total amount of erosion increased by over half, the erosive power of a gram of powder remains practically constant, as it did in the experiments at constant pressure.

This series of experiments confirms the former and shows that after reaching a pressure in the neighborhood of 2,000 kilograms, which is sufficient to give the escaping gas a high

eroding efficiency, a further increase in pressure plays but a secondary part, and that the volume of escaping gas is the most important eroding element.

VI. Effect of the Rate of Combustion.

Until now we have assumed as one of the necessary conditions in order to attain reliable results as to different forms and methods of erosion, that the rate of escape of gas through the axial hole be not affected by the rate of burning of the powder; and in order to fulfill this condition we have used relatively quick small-arm powders with which the time of reaching maximum pressure after ignition is so short that the amount of gas escaping up to that time may be neglected. We know that this condition is practically fulfilled with the BF–AM and VF powders, since the pressure obtained when using the pierced specimen does not differ from that obtained in a closed chamber of equal capacity. It still remains to be shown within what limits of quickness the above conditions hold, and to ascertain the effect on the amount of erosion of the rate of combustion.

A powder of certain composition can be formed into pieces of varying size with very different methods of ignition, of which, under the conditions of our experiments, the time of attainment of maximum pressure would vary from a few $\frac{1}{10000}$ths of a second to a few $\frac{1}{100}$ths of a second.

The rate of combustion is registered in each case upon a revolving cylinder by means of a crusher or spring gauge.

By the crusher gauge we obtain the following:

(1) The time of the hang-fire which elapses between the ignition of the primer and the first compression of the gauge, corresponding to a pressure of about 150 kilograms.

(2) The time of compression, which is nearly equal to the time of combustion in powders that are not too quick, but which is greater in every case than the time of combustion.

The spring gauge registers not only the rate of increasing pressures but the curve of decreasing pressures as the gas escapes.

In an earlier series of experiments I have compared the erosive powers and the rates of combustion given by different powders of the same composition, nitrocellulose colloided by some solvent. These were of greatly different quickness.

In the following experiments—

Powder T–bis is a flake powder, used in revolvers.

Powder BF–AM is a slower powder, used in service small arms.

Powder BM–7, always of the same type, is used in the French naval guns of 164-millimeter caliber (6.46 inches).

Powder BM–13, intended for naval guns of 320 and 340-millimeter caliber (12.60 and 13.39 inches), is one of the slowest that can be employed.

QUICK POWDERS.

The following table contains results of experiments obtained with the first three powders:

Powder.	Charge.	Press. per sq. cm.	Eros. mm^3.	Eros. per gram.	Results obtained from the curves.
T–bis....	3.55	<2,528	26.4	7.4	Hang-fire, 2 ms. Dynamic compression in $\frac{8}{10000}$ of a second.
T.....	3.55	<2,588	23.1	6.5	Hang-fire, 2 ms. Dynamic compression in $\frac{5}{10000}$ of a second.
BF–AM.	3.55	2,300	23.0	6.4	Hang-fire, 4 ms. Static compression in 2.4 ms. $\frac{dp}{dt}M = 1{,}206$ t.

The normal pressure in a closed chamber with this kind of powder and a density of loading of .20 is 2,300 kilograms.

The slightly greater results obtained with the very quick powders T–bis and T are due to the dynamic action of the pressure apparatus, and it may be said for all these powders that the initial flow of gas does not affect the maximum pressure.

An examination of the curves obtained confirms this result.

Also the maximum value of dp/dt, 1,206 tons obtained for the BF–AM powder with a pierced test specimen, is practically the same as the value, 1,237 tons per second, obtained with the same powder in a closed chamber.

Under these conditions it is seen that the erosive power remains nearly constant, although for the powders used the rates of increase of pressure vary greatly, the compression of the gauge taking five or six times longer for the BF powder than for the T and T–bis powders, even with twice the time of ignition—4 instead of 2-thousandths of a second. (In the tables ms. stands for mil-second.)

SLOW POWDERS.

Passing from the foregoing quick powders to powders of the same composition but of slow combustion, the effect of the initial flow of gas is immediately seen, even when an attempt is made to reduce its duration and relative importance by increasing the charge and pressure.

Powder.	Charge.	Press. per sq. cm.	Eros., in mm^3.	Eros. per gram, mm^3.	Results from curve.
BF–AM.....	5.0 g.	3,614	40.0	8.0	Hang-fire, 1 ms.; time of combustion, 1.25 ms. $\frac{dp}{dt}M = 3{,}426$ tons.
	5.0 g.	3,688	35.8	7.2	Hang-fire, not taken. Experiment made with axial hole closed with a lead wire.
BM–7AM ...	5.0 g.	2,694	26.0	5.2	Hang-fire, not taken; time of combustion, 7 ms. $\frac{dp}{dt}M = 605.7$ tons.
BM–13AM..	5.0 g.	2,012	17.2	3.44	Hang-fire, 10 ms.; time of combustion, 14 ms. $\frac{dp}{dt}M = 150$ tons.
	5.0 g.	2,320	27.6	5.52	The second experiment gave with the spring gauge for the sum of the times of ignition and combustion, 20 ms.

The preceding table gives the results of experiments with such powders:

The pressure in a closed chamber with the same density of loading was found to be 3,698 kilograms for the BF–AM powder, and 3,593 kilograms for the BM–7 powder. The BM–13 powder is about the same. A reduction of 1,000 to 1,500 kilograms in the values of the observed maximum pressures with a pierced test specimen shows that in the case of slow powders the initial flow of gas assumes a considerable importance; at the same time, the erosive power decreases and varies according to the conditions of combustion. The results of an analysis of the curves supply the reasons.

It was seen from the curves of the spring gauge that the maximum pressure of about 2,300 kilograms, obtained either with a charge of 3.55 grams of BF powder or with a charge of 5 grams of BM–13 powder, gave exactly identical curves of decrease in rate of pressure after the maximum. Also the pressure of 2,300 kilograms is lowered for the two powders to 1,000 kilograms, below which we have seen that the erosive power is very slight, in times of from 17 to 19 ms.; this pressure of 2,300 kilograms is lowered to 500 kilograms in 35 to 37 ms. But the law of increase of pressure in the two cases is entirely different, as is shown below:

Powder.	Hang-fire.	Time of combustion.	Max. value of $\frac{dp}{dt}$.
BF–AM	4 ms.	2.4 ms.	1,206 tons.
BM–13	10 ms.	14.0 ms.	150 tons.

It is, therefore, seen that the escape of gas under the low initial pressures which are inefficient from an erosive point of view, assumes considerable importance in the case of a slow powder, and greatly reduces the erosive power of the charge.

In brief, these relative experiments with powders of the BF type prove that, under the condition that the initial flow of gas does not appreciably reduce the value of the maximum pressure below that obtained in a closed chamber when using the same density of loading, the resultant erosion is charac-

teristic of the type of powder under experiment, and within large limits does not depend upon its rate of combustion.

I have attempted to extend this conclusion relating to powders of the pure nitrocellulose type to powders of the ballistite type containing 50 per cent. of nitroglycerine.

A sample of powder much quicker than VF powder was prepared by cutting the grains much smaller. The number of grains per gram thus varied from 99 to 950.

The following table permits a comparison of the erosive power and the rate of combustion of these two powders, using the same charge in the 17.8 cubic-centimeter bomb and recording the increase in pressure by means of the registering crusher-gauge apparatus.

Powder.	Charge.	Press.	Eros.	Eros. per gram.	Results from the curves.
VF-fine ...	3.55	2,597	83.6	23.5	Hang-fire, 1.2 ms.; time of combustion, 0.9 ms. $\frac{dp}{dt}M = 4{,}604$ tons.
VF	3.55	2.416	84.5	23.8	Hang-fire, 1 ms.; time of combustion, 2.4 ms. $\frac{dp}{dt}M = 1{,}547$ tons.

The observed pressures do not differ from the normal pressure in a closed chamber for this type of powder, which is about 2,500 kilograms for this density of loading.

From these results it is seen that the quickness of the powder may vary within the ratio of 1 to 3 without altering the amount of erosion.

The deduction drawn above in the case of the BF powders seems, therefore, to be sufficiently correct to be considered as a general law.

PRACTICAL SIGNIFICANCE OF THE EXPERIMENTS.

After having determined the conditions under which the proposed experiments are to be conducted, it now remains to find out what may be their practical value from the point of

view of preventing the destructive effect of erosion in guns. After what we have said of the character of ballistic erosion, that which is brought into the question by the gun and that produced in the experiments are undoubtedly of the same class; they only differ in amount—the amount of escaping gas and the time of escape being far greater in proportion in the case of the experiment, thus allowing a single firing to produce measurable results.

The velocity of the projectile in the region corresponding to the maximum pressure, where the erosion is the greatest, is at least 100 to 150 meters per second. A longitudinal strip 1 centimeter in length is, therefore, in contact with the rotating band during a period of about $\frac{1}{10000}$ of a second. On the other hand, we have found from our experiments that the time of escape under a pressure varying from 2,500 to 1,000 kilograms reaches $\frac{17}{10000}$ to $\frac{19}{10000}$ of a second, a period 200 times greater.

We have seen in the experiments that the erosion within large limits is proportional to the amount of escaping gas when under the same maximum pressure, and that for the same amount of gas the pressure exerts but a secondary influence after it has reached the neighborhood of 2,000 kilograms.

In the gun the escape of each elementary jet of gas is accomplished under practically the same pressure, on account of the very short time of its existence; it will, accordingly, be proportional to the amount of leakage, or, in other words, to the amount of pressure, which does not cause erosion directly, but indirectly, through the effect it has on the amount of leakage of gas.

A comment of the same nature also applies to the effect of the charge on ballistic erosion. The amount of the charge ought not to cause erosion directly, because it is always vastly greater in proportion than the amount of leakage of an elementary gas jet. But it may be conceived that it assists through the action it exerts on the rate of increase of the pressure and the movement of the projectile.

When the charge of the same powder is increased in a gun, at every point in the travel of the shell along the bore the pressure is raised in the same proportion as the velocity of the rotating band is increased, and these two influences would seem to compensate each other from the point of view of the escape of gas and the erosive effect. But it is easily seen that it is the effect of the pressure that counts, and that ballistic erosion ought to increase with an increase in charge. This is an observed result from experiments.

When the charge of a gun is increased by using slower powders without changing the maximum pressure, the ballistic formulae of M. Sarrau show that the point of bore reached by the projectile at the moment of maximum pressure is moved towards the muzzle, and this fact ought to carry the erosion beyond to a portion of the rifling which was immune with quicker powders. This also is a result proved by experience.

On the whole, everything leads to the belief that the classification of the metals and powders according to their susceptibility to erosion determined by the tests which have been described, may be valuable to assist in providing against the results of ballistic erosion, and it was with this object that the experiments have been undertaken.

CHAPTER II.

The method of experimentation described in the preceding chapter permits us to take up the study of the effect on different metals of the erosion caused by the escape at high pressure of the gases of an explosive.

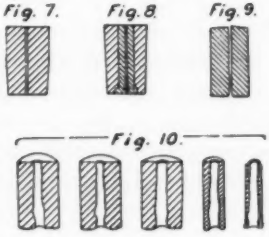
METALS OF THE IRON GROUP.

The greatest number of experiments have been made with iron and its alloys, as that is the only metal possessing a practical interest from the point of gun construction.

In the following table I have given a list of the metals of this class with which experiments have been made, also indi-

cating from whom the specimens were received, their chemical composition and the physical properties of certain ones.

I owe my warmest thanks to the Director of the Marine Laboratory, to M. Le Chatelier, to M. Osmond and to M. Charpy, who have placed at my disposition, with careful descriptions, a large number of varied specimens of such materials as might be considered interesting from the point of view of the study which I have undertaken.



Experiments were made with the larger part of the specimens in the form of a cylinder 22 millimeters in diameter and 40 millimeters long (See Fig. 7). Some having an insufficient diameter were fitted in another specimen with the above dimensions, a copper gas check being added as shown in Fig. 8.

The susceptibility to erosion of these different metals was determined in the experiments with the VF powder, or ballistite type, with 50 per cent. of nitroglycerine, which possesses the greatest erosive power of the ordinary powders and would be the best one to accentuate the difference of their erosion resistance of the above metals.

A few of the experiments were made with the large 343 cubic-centimeter bomb with charges of 70 grams, but the greater share were made with the small 17 cubic-centimeter bomb with charges of 3.50 grams, which was found from the preceding experiments to be the best.

LIST OF METALS.

Metal.	From whom received.	Car.	Chro.	Si.	Tung.	Ni.	Elas. str.	Tens. str.	Elong., per cent.
Fer au bois, BFM.....	Lab. Mar.	...	...	...	...	...	...	...	...
Swedish iron.....	"	...	...	...	...	...	...	...	...
Mild steel.....	"	<.01	...	...	...	...	29	48	24.0
Gun steel.....	"	<.01	...	...	...	...	46	72	15.0
Chrome tool steel:									
Untempered	"	...	...	...	...	...	...	...	...
Tempered.....	"	...	...	...	...	...	...	118	0.6
							Mang.		
Chrome steel, C-1.....	M. Le C.	.50	2.5	.27	...	...	.23	...	...
C-3.....	"	1.07	3.5	.36	...	...	.21	...	...
Tungsteen steel, V-1...	"	.55	...	.20	2.9	...	.42	...	...
V-3...	"	1.10	...	.32	2.7	...	.38	...	...
Chrome steel, ECR.....	M. Char.	.12	2.0	...	...	...	...	...	...
6C.....	"	.40	2.0	...	...	...	...	...	...
Steel, CF-10.....	"	.35	1.0	...	...	2.0	...	...	...
KF-9.....	"	.35	2.0	...	...	4.0	...	...	...
Annealed steel, No...						13 to			
766	M. Osm.	.259	.59	...	...	14	.43	...	...
Same, not annealed.....	"	...	...	...	...	...	...	...	...
Annealed steel, No...									
836.....	"	.255	...	...	...	24.4	.44	...	...
Same, not annealed....	"	...	...	...	...	...	...	...	...
Steel, No. 851.....	"	.070	...	...	...	26.64	.285	...	...
850.....	"	.104	...	...	...	43.44	.388	...	..
Soft cast iron.....	"	...	...	...	...	...	...	...	...

The observed erosions, reduced to cubic millimeters, are given in the following table:

TABLE I.

343 cubic-centimeter bomb. Charge, 70 grams. Powder, VF. Density of loading, .20.

Kind of metal.	Observed erosion.
Mild steel, ...	1,055 mm³.
Gun steel, ...	1,014
Chrome tool steel, tempered, . . .	1,125
Chrome tool steel, not tempered, . .	1,088
Tungsten steel, V-1, ...	1,055

TABLE II.

17.8 cubic-centimeter bomb. Charge, 3.55 grams. Powder, VF. Density of loading, .20. Pressures measured by crusher gauge.

Kind of metal.	Erosion.	Max. pressure.
BFM iron	68.2 mm³.	2,474
Swedish iron	75.0	2,602
ECR steel (2 per cent. chromium, 0.10 per ct. carbon)	75.5	2,509
Soft steel, No. 1	78.7	2,407
Gun steel	84.5	2,509
Gun steel	83.2	2,370
Gun steel	83.6	2,597
Steel, C-6 (2 per cent. chromium, 0.40 per ct. carbon)	83.8	2,504
CF-10 (1 per ct. chromium, 2.0 per ct. nickel)	83.97	2,625
C-1 (2.5 per cent. chromium)	88.0	2,430
RF-9 (2.0 per cent. chromium, 4.0 per ct. nickel)	89.5	2,504
V-1 (2.7 per cent. tungsten)	95.1	2,333
No. 766, untempered (13 to 14 per cent. nickel)	95.1	2,467
C-3 (3.5 per cent. chromium)	98.5	2,509
Pure nickel	104.6	...
Steel, No. 766, tempered (13 to 14 per cent. nickel)	106.7	2,375
No. 836, tempered (24 per cent. nickel)	107.1	2,430
not tempered (24 per cent. nickel)	111.5	2,509
No. 851 (26.64 per cent. nickel)	116.9	2,453
Soft cast iron	131.0	2,532
Steel, No. 850 (43.44 per cent. nickel)	134.9	2,485

These results lead to the following comments:

(1) Of all the metals and alloys of the iron group, pure iron and steels with a very small percentage of carbon give the least erosion. Small quantities of chromium, manganese, nickel and tungsten, at least when combined with .40 to 1.00 per cent. of carbon, are of no assistance in resisting erosion. Large percentages of nickel greatly increase the amount of erosion, the richer alloys even exceeding pure nickel in the amount of resulting erosion.

(2) The effects of tempering, and other physical properties acquired at an ordinary temperature by suitable treatment, are practically nil in reducing erosion.

(3) The classification of metals according to their susceptibility to erosion when using the same powder seems to depend on their temperatures of fusion. Thus it is seen that

pure iron, with a melting point of about 1,600 degrees, is at the top of the list, followed by steel with a low percentage of carbon, then by steels with increasing proportions of carbon, and, finally, at the other end of the list is soft cast iron, with a melting point of about 1,220 degrees.

I will return to this question of temperature later in discussing the physical properties of erosion.

OTHER METALS AND ALLOYS.

It has appeared useful to extend the experiments to metals other than those of the iron group, less with a view as to any practical information as to their use in gun construction, such as adding a surface deposit of some metal, but with the object of demonstrating in as clear a manner as possible the elements which enter to the great extent in the phenomenon of erosion.

In these relative comparisons of metals of different densities it is necessary to express the amount of erosion in volume, as has been done in most of the preceding experiments.

The following metals have been used:

Pure platinum.
Platinum iridium.
Copper of different kinds.
Pure silver.
Alloy of silver and copper.
Bronze.
Brass.
Zinc.
Aluminum.

The following table contains the results obtained with these metals, and also pure iron, for the purpose of comparison, using the 17.4 cubic-centimeter bomb with a charge of BF powder giving a maximum pressure of about 2,500 kilograms per square centimeter.

Kind of metal.	Specific weight.	Erosion in mm³.	Pressure.	Melting Point
Pure platinum	21.5	59.1	...	1,800
Platinum iridium	21.5	73.7	...	...
Commercial copper	8.9	90.8	...	1,054
	8.9	61.6	...	1,054
	8.9	79.7	...	1,054
	8.9	48.7	...	1,054
Copper for crusher-gauge cylinders	...	98.8	2,426	...
	...	100.0	2,593	...
Pure silver	10.53	230.8	2,588	954
Silver alloy, $\frac{997}{1000}$	10.50	268.4	2,511	...
Bronze	...	278.8	2,546	...
Brass	8.40	326.0	2,578	1,015
Zinc	7.15	1,017.9	2,462	412
Aluminum	2.60	2,169.0	2,044	625
	2.60	2,307.0	2,059	625
Pure iron	7.80	68.2	2,474	1,600

If the pure metals are considered, excluding the alloys of bronze and brass, it is seen that the erosion increases in inverse ratio to the melting point, with the exception of aluminum, which gives twice as much erosion as zinc, although its melting point is considerably greater. It may be stated that this metal when near its melting point is subject to a certain disintegration which undoubtedly plays an important part in the ease with which the heated layers of metal are torn away by the escaping jet of gas. It is also undoubtedly due to the peculiar molecular formation of certain alloys, such as brass, which have been found to exist through the researches of M. Charpy, that a greater susceptibility to erosion is shown by such alloys than would be expected from their temperatures of fusion.

Pure platinum gave the least volume of erosion, but its superiority is small when compared to pure iron, and if the weight of metal eroded is considered its loss is two or three times greater than that of the latter.

Copper offers about the same amount of resistance to erosion as the different steels, and on account of the ease with which firing vents may be repaired with this metal it is generally used for this purpose.

In brief, these comparative experiments preclude any hope of discovering in the iron group of metals, or in other metals, either rare or common, any substance which possesses greater resistance to erosion than do the low-carbon steels at present in general use for the construction of guns of large and small caliber.

It is in the explosive and not in the metal that we must search for the remedy to prevent the erosion in guns.

CHAPTER III.

EFFECT OF THE TYPE OF EXPLOSIVE ON THE AMOUNT OF EROSION.

The preliminary experiments made to determine the limiting conditions show plainly the great influence exerted on the amount of erosion by the type of explosive.

These experiments have been extended to include all the principal ballistic powders in use in Europe, and also the ordinary explosives.

In every case the samples were prepared as quick powders, in order to fiulfill the fundamental condition established in Chapter I, that the maximum pressure obtained when using the pierced-test specimen must always be in the neighborhood of that obtained in a closed chamber with the same density of loading.

Most of the experiments were made in the 17.8 cubic-centimeter bomb, using specimens of gun steel. A few were also made in the 75 cubic-centimeter bomb and the 343 cubic-centimeter bomb.

Colloids of Pure Nitrocellulose.

Experiments were made with the following powders of varying quickness:

T–bis, . .	Revolver powder.
T, . . .	Shot-gun powder.
BF, . . .	Service small-arm powder.

Colloids with a Nitroglycerine Base.

The following powders were used:

VF–bis,	. .	A very quick revolver powder.
VF,	. .	Powder for service small arms.

These powders (VF–bis and VF) are of the ballistite type, containing 50 per cent. of nitroglycerine and 50 per cent. of nitrocellulose of high nitration, colloided with acetic ether.

CF=service small-arm powder of English cordite type, containing 57 per cent. of nitroglycerine, 5 per cent. of vaseline and 38 per cent. of high-nitration nitrocellulose colloided with acetone.

Italian solenite=small-arm powder containing 34 per cent. of nitroglycerine, 63 per cent. of nitrocellulose and 3 per cent. of hydrocarbons.

Black Powders.

The following powders were used:

Sporting powder, No. 4, of French manufacture, containing 78 per cent. of saltpeter, 10 per cent. of sulphur and 10 per cent. of carbon.

Rifle powder, No. F3, containing 75 per cent. of saltpeter, 10 per cent. of sulphur and 15 per cent. of carbon.

Different Explosives.

Gelatine dynamite, of Cugny, containing 94 per cent. of nitroglycerine and 6 per cent. of soluble nitrocellulose.

"Dynamite de Vouges"=75 per cent. of nitroglycerine and 25 per cent. of Kieselguhr.

Nitromannite, a nitric ether of the hexatonic alcohol, mannite ($C_6H_{14}O_6$), granulated to form a quick powder.

Nitroguanidine, powdered and carefully mixed in such a manner as to facilitate the combustion of the naturally slow-burning substance.

The derivatives of guanidine have been prepared commercially for more than ten years, especially in Germany. The

nitrate of guanidine, or nitroguanidine, was introduced into France for use in the War Department, in 1896, by MM. Pelliot and Hoffman, after improvements in its manufacture had been made in Germany by Dr. Flemming. The nitroguanidine has been proposed especially for the purpose of giving stability to the nitrocellulose and of lowering the temperature of combustion. In France the manufacture and properties of derivatives of guanidine have been the object of extended experiments by M. Patart (See "Zeitschrift für Angewandte Chemie," t. XLVI, 1898).

The following table gives the results of the experiments, using a gun-steel test specimen 40 millimeters long, with an axial hole 1 millimeter in diameter:

Powder.	Charge.	Pressure.	Erosion, mm^3.	Erosion, per gram.	Force.	Temperatures.
T-bis	3.55	2,528	26.4	7.4	9,600	2,676° (Result of experiments with closed chamber at Laboratoire Central.)
T	3.55	2,588	23.1	6.5		
BF	3.45	2,403	20.3	5.88		
	3.50	2,361	22.7	6.58		
	3.55	2,224	24.7	6.96		
	3.55	2,253	25.5	7.19		
	3.55	2,143	20.1	5.66		
VF-bis	3.55	2,597	83.6	23.5	10,000	3,384° (Experiments at Laboratoire Central.)
VF	3.47	2,509	84.5	24.3		
	3.51	2,370	83.2	23.7		
	3.55	2,542	90.2	25.4		
	3.55	2,360	85.9	24.2		
	3.55	2,416	84.5	23.8		
Cordite	3.55	2,500	64.2	18.1	10,000	
Solenite	3.55	2,449	48.9	13.8	10,000	
Powder, F-3	10.00	2,167	22.3	2.2	3,000	2,910° (M. Sarrau.)
Sporting powder	8.88	1,958	40.0	4.5	3,000	3,530° "
Gel. Dyn	3.35	2,458	105.0	31.4	10,000	3,545° "
Dyn., 75 per cent.	3.35	2,084	60.0	18.0	8,000	3,161° "
Nitromannite	3.54	2,361	83.5	23.6	10,000	3,429° "
Nitroguanidine	3.90	2,019	8.8	2.3	9,000	907° (M. Patart.)

(1) From the above it is seen that with ballistic powders of about the same force the one with the greatest temperature of combustion gives the greatest amount of erosion.

The VF powders of the ballistite type have about four

times the erosive power of nitrocellulose powders, but their temperature of combustion is about 700 degrees greater than that of the latter.

English cordite, in spite of the high percentage of nitroglycerine (57 per cent.), has an erosive power less than that of ballistite, but from one-half to three times greater than that of pure nitrocellulose. The temperature of combustion is reduced in this powder by the addition of 5 per cent. of vaseline.

The Italian solenite, in which the percentage of nitroglycerine is reduced to 34 per cent., is still twice as erosive as the BF powders.

Finally, for the nitroguanidine having a temperature of combustion of about 1,000 degrees the erosive power is lowered to one-third of that of the BF powders. This great decrease may be attributed in a slight degree to its force, which is rather low, and to its slow combustion.

(2) In passing from ballistic powders to high explosives of the same force we find that those with the highest temperature of combustion have the greatest power of erosion.

The gelatine dynamite, which is the strongest and has the greatest heat of combustion, is the most erosive of the whole series.

(3) The results obtained from the group of black powders cannot be compared directly with those of the stronger powders. The ratio of forces is about 1 to 3, so that in the black powders the density of the jet of gas escaping under the same pressure is almost three times the density of the resulting gas of the B powders or of ballistite. Since the amount of escape of gas varies as the square root of the density it will be necessary to almost double the erosive coefficients per gram of the black powder in order to compare them under conditions of similar ballistic action with the other powders.

But if the erosive coefficients of the black powders are compared among themselves this difficulty disappears, and we have a striking confirmation of the influence of the temperature on the amount of erosion.

The F–3 powder and the sporting powder differ in the proportion of their ingredients, 75–10–15 for the first and 78–10–12 for the second. Their forces are very similar. But the quantity of heat given off by the F–3 powder is much smaller and the volume of gas much greater than the similar elements of the sporting powder. This point has been established through the researches of Nobel and Abel. The temperatures of combustion calculated by M. Sarrau, according to the thermo-chemical data of M. Berthelot, are 2,910 degrees and 3,530 degrees, respectively.

In other words, the erosive coefficient of these powders of the same type and ballistically equivalent is doubled by the foregoing increase in temperature of combustion.

75 CUBIC-CENTIMETER BOMB. SPECIMENS OF GUN STEEL.

Kind of powder.	Charge.	Erosion, in mm^3.	Erosion per gram.	Temperature, in degrees.
BF	14.66	109.2	7.45	2,676
BF	14.66	105.0	7.30	2,676
VF	14.66	322.0	21.96	3,384
Pure nitroguanidine	18.11	41.5	2.29	907

343 CUBIC-CENTIMETER BOMB. SPECIMENS OF GUN STEEL.

BF	69.08	383.0	5.54	2,676
BF	69.17	392.0	5.67	...
BF	68.99	374.0	5.42	...
Cordite	69.17	738.0	10.70	...
VF	69.17	1,014.0	14.70	3,384
F-3 powder	172.5	427.0	2.47	2,910
Sporting powder	172.5	715.4	4.15	3,530

Thus it is seen that while the widest variation in the composition of metal suitable for gun construction has given us only minor changes in the erosive coefficient, by changing the type of explosive with practically the same power we can vary the erosive power of these explosives in the ratio 1 to 10 or 15, and it is the temperature of combustion which seems to be the cause of such a variation.

I have attempted to confirm this conclusion by further experiments made under the same maximum pressure but with much greater charges in the 75 and the 343 cubic-centimeter bomb. The results obtained are given in the preceding table.

This new series confirms in all their essential points the conclusions drawn from the first series of experiments.

The classification of the BF powders, cordite and VF remains the same and, with their ratios of erosive powers, in the same order as those given by the 17 cubic-centimeter bomb.

In the 75 cubic-centimeter bomb, with a charge five times as great, the nitroguanidine still maintains an extremely low erosive coefficient, equal to about one-third that of the BF powders.

Finally, the two types of black powder gave the same results in the 343 cubic-centimeter bomb as they did in the 17.8 cubic-centimeter bomb with a charge twenty times smaller—an erosive coefficient that doubled for an increase of 600 degrees in the temperature of combustion.

The difference in the absolute values of the erosive coefficients of the same powder, obtained in the different bombs, is due to the increase in diameter of the axial hole and consequent reduction in the amount of metal eroded, as has been explained in Chapter I.

Methods of Reducing the Erosive Power of Existing Powders.

The preceding experiments permit us to predict that by the combination of very erosive powders with a material which is slightly erosive, such as nitroguanidine, a product may be obtained with an intermediary erosive power which may be varied at will by the proportions of the component parts.

I have experimented along this line with different mixtures of B powders, cordite, and ballistite with nitroguanidine.

In this class of experiments where the samples of powders used are prepared in the laboratory in limited quantities, a method of experiment limiting the charges to small quantities is most necessary.

The following table gives the erosive power of these different samples, prepared by mixing and pressing, in the form of the quick small-arm powders:

Form of sample.	Charge, grams.	Pressure, kg.	Erosion, mm^3.	Erosion, per gram.
BF	3.52	2,277	20.7	6.4
Mixture { 70 per cent. BF powder / 30 per cent. nitroguanidine...	3.74	2,407	20.4	5.45
Mixture { 50 per cent. BF powder / 50 per cent. nitroguanidine...	3.43	2,124	11.8	3.44
Cordite	3.55	2,500	64.2	18.1
Mixture { 70 per cent. cordite / 30 per cent. nitroguanidine...	3.55	2,407	46.2	13.0
Mixture { 50 per cent. cordite	3.55	2,244	26.1	7.37
50 per cent. nitroguanidine....	3.55	2,375	30.0	8.44
VF-ballistite	3.53	2,466	85.3	24.3
Mixture { 70 per cent. ballistite / 30 per cent. nitroguanidine....	3.55	2,706	50.4	14.2
Mixture { 50 per cent. ballistite / 50 per cent. nitroguanidine....	3.55	2.435	27.8	7.8

From the above it is seen that the incorporation of 50 per cent. of nitroguanidine with powders of the cordite and ballistite types reduces their erosive powers to those of the BF powder class. The same treatment applied to the BF powders would reduce their erosive power, already very low, by one-half, if such a reduction were considered necessary, as, for example, in the case of ammunition for automatic guns where a large number of shots are fired in a short space of time.

The accuracy of the above results were checked with much larger charges in the 75 cubic-centimeter bomb. The following table gives the erosion observed with the BF powder, and with a mixture of this powder and nitroguanidine:

75 CUBIC-CENTIMETER BOMB. SPECIMEN OF GUN STEEL.

Kind of powder.		Charge.	Erosion.	Erosion, per gram.
BF		14.66	107.1	7.4
Mixture	50 per cent. BF powder	14.79	77.0	5.2
	50 per cent. nitroguanidine	14.79	68.6	4.6

The decrease in erosive power, with five times as great a charge, remains in the same proportion as in the experiments made with a small charge in the 17.8 cubic-centimeter bomb.

In brief, we have established in this chapter that the temperature of combustion of the explosive plays a preponderating part in the phenomenon.

By the combination of explosives having different erosive power the erosive power of the mixture is reduced to some intermediate value.

Even today in the artillery are powders of similar force, but whose erosive powers vary from 1 to 10.

CHAPTER IV.

PHYSICAL CHARACTERISTICS OF EROSION.

From the preceding experiments a few conclusions relative to the physical characteristics of erosion may be drawn, in addition to the practical and ballistic results already given.

We have seen that this phenomenon requires for its production that the escape of gas takes place under considerable initial pressure, with the resultant high velocity of the jet of gas. The erosion produced by the same mass of gas at a high temperature is practically nothing at pressures under 100 atmospheres, but increases rapidly as the pressure increases, and finally reaches a limit at about 2,000 kilograms. From this point upwards, even to 4,000 kilograms, the increase in the amount of erosion is hardly appreciable.

Erosion depends first, therefore, on a certain very slight

physical action for its production. A second element, much more important, assists in producing it—that is, the melting point of the metal composing the canal of escape and the temperature of the jet of gas.

Thus with colloids of the ballistite type, with its 50 per cent. of nitroglycerine, or the BF type of pure nitrocellulose, the erosion increases in the ratio of 1 to 15 or 20 when using zinc in place of gun steel.

On the other hand, the erosion of the same metal, as gun steel, for instance, is doubled by substituting for an explosive with a low temperature of combustion, such as nitroguanidine (1,000 degrees), one of the ballistite type of which the temperature of combustion reaches about 3,400 degrees.

This leads us to believe that ballistic erosion is the result of a compound action:

(1) The softening of the metal under the action of a jet of gas of high temperature, incessantly renewed, and capable of furnishing, in spite of its low specific heat, the amount of heat necessary to raise the temperature of the metal and change its condition. It can thus be explained why the remarkable physical properties of certain metals or alloys, acquired by tempering or annealing, are of no effect in preventing erosion, because these differences in structure are nullified in the preparatory stage of liquefaction.

(2) A physical action sufficient to eject the softened or molten outside layer of the hole through which the gas is escaping.

The conjunction of these two actions is indispensable. In a closed chamber, or with gas escaping at a low pressure, the physical action above described does not take place, and the combustion of explosives with the greatest heat does not produce erosion.

Inversely, explosives with a very low temperature of combustion produce but negligible erosions, even under the highest pressures, because the powerful physical action of the gas produces no effect without the simultaneous softening of the surface by heat.

In brief, ballistic erosion may be compared to the blowing away a film of liquid from the inside surface of a small pipe by the breath of the operator.

An examination of the physical characteristics produced by erosion on metallic test specimens upholds this theory.

PHYSICAL CHARACTERISTICS OF THE ERODED SPECIMENS.

A metallic specimen 40 millimeters long, with an axial hole 1 millimeter in diameter, has the following characteristics after erosion:

If the erosion is small the hole remains practically cylindrical; at the end towards the charge the hole is widened slightly, as indicated in Sketch No. 9.

On the other end a small conical boss is raised on the smooth surface of the specimen. It is formed of a mass of globules which are clearly seen under the microscope, forming a sort of cone of ejection, loosely combined, and which may be scraped off with the finger nail or a piece of wood.

It is probable that this deposit is formed in the latter phase of the escape of gas when the pressure is too small to eject to any distance the liquid or pasty fragments of metal.

If the erosion is increased by increasing the charge the widening of the hole on the powder-chamber side is increased until it resembles the bell of a French horn. The streams of gas then flow into the hole with an increasing convergence, and it loses its cylindrical form to take that of a string of ovoids separated by a series of equidistant contracted sections, as shown in Fig. No. 10.

These figures have some analogy with those obtained by photography of the escape of streams of a liquid, but it is not probable that the longitudinal vibratory movements of the stream of gas would have any influence.

It is more reasonable to suppose that the initial convergence of the streams increasing with the erosion at the chamber end of the hole determine the successive reflexions of the streams of gas on the walls. From thence these points are

the special object of attacks by heat, and consequent maximum erosion.

The above are the normal characteristics observed in every case where the metal is perfectly homogeneous.

But it is not uncommon in alloys, and even in the pure metal when there is some lack of homogeneity, to see an irregular form of erosion. In places along the axial hole may be seen abrupt projections against which the stream of gas is broken into eddies, resulting in the formation of irregular pockets. See Fig. 11.

Fig. 11.

The extreme erosion obtained with specimens of aluminum presents some peculiar characteristics. In place of the series of ovoids scattered along the hole, there is but a single cavity beyond the contraction at the powder end of the specimen, and this cavity extends the whole length of the specimen, which is reduced to a thin metallic wall. See Fig. 12.

Fig. 12.

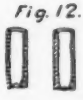


Fig. 13.

The gas escaping from the axial hole still possesses considerable energy. It escapes through a threaded plug of gun steel which holds the specimen in place, and has a cylindrical hole in the outer end 10 millimeters in diameter and 30 millimeters long. Although the area of this hole is one hundred times greater than that of the original axial hole of the specimen, the walls of this plug were deeply eroded by heavy charges, as is shown in Fig. 13.

Fig. 14.

ANALYSIS OF THE PHENOMENON OF EROSION.

Setting aside irregularities which may be attributed to lack of homogeneity of the metal, the process of cylindrical erosion may be compared to the process propagation of heat through a homogeneous body which is changing its condition. But it offers a certain complication: On account of the progressive elimination by the jet of gas of the particles of metal which have reached the disintegrating temperature, the surface to be reheated is constantly being carried outward by the constant removal of the surface film of metal by the incandescent stream of gas. This surface to be reheated travels, therefore, at the same velocity as that of the erosion. These conditions may be expressed by the following analytical equations:

Assume at any instant of time, t, the cylindrical surface of radius r which has just been stripped of its molten film by a jet of gas. This surface is at the temperature of disintegration, θ_d. It is in contact with gases at a temperature of explosion, θ_e. On its right side this wall receives per unit of surface in the time dt a quantity of heat proportional to the difference of temperatures, $\theta_e - \theta_d$, and to a coefficient of exterior conductivity, h. On its left side this wall loses a quantity of heat—

$$K\left(\frac{d\theta}{dr}\right)\theta_d = h(\theta_e - \theta_d), \tag{1}$$

which does not correspond to a fixed abscissa, but to a determined value of $\theta = \theta_d$.

For a large value of r we always have

$$\theta = 0, \tag{2}$$

which forms a second relation to the surface.

In the beginning the cylindrical surface of radius r of the axial hole is at a temperature θ_d.

The quantity of heat given to a film of metal of thickness dr is obtained by taking the difference of the flow of heat which takes place during the time dt across the limits of the

cylindrical surfaces, and it will be assumed that this quantity of heat serves to raise the temperature of the film.

The flow of heat at right angles to a surface of radius r and height equal to unity is

$$-2\pi r K \frac{d\theta}{dr} dt.$$

The flow at right angles to a surface of radius $r + dr$ is

$$-2\pi r(r + dr) K \left(\frac{d\theta}{dr} + \frac{d^2\theta}{dr^2} dr\right) dt.$$

The difference of flow is by omitting the negligible higher powers:

$$2\pi r \left(\frac{d^2\theta}{dr^2} + \frac{1}{r} \cdot \frac{d\theta}{dr}\right) K dr dt.$$

The quantity of heat absorbed by the film to raise its temperature $d\theta$ is

$$2\pi r \, dr \partial c \frac{d\theta}{dt} dt,$$

where ∂ indicates the density of the material and c a calorific capacity which will include all the simultaneous transformations which accompany a rise in temperature.

By equating the difference of the flow to the quantity of heat absorbed, the above becomes—

$$K \left(\frac{d^2\theta}{dr^2} + \frac{1}{r} \cdot \frac{d\theta}{dr}\right) = c\partial \frac{d\theta}{dt}. \tag{3}$$

With this equation in conjunction with equation 1, we may determine the function $\theta = \varphi(r, t)$ where $\psi = \delta, t)$.

The value $r = \varphi(\theta, d, t)$ then gives the radius of the eroded hole as a function of the time.

Without attempting the complete analytical solution of this problem, we may venture the following statements:

(1) The quantity of heat transmitted to the surface depends on the excess of the temperature of the gas over that disinte-

grating temperature of the metal, according to the preponderating effect of the temperature of combustion of the explosive, or of the melting point of the metal, as shown in the experiments. But the amount of heat transmitted is not the sole factor, but is affected by the relative values of the coefficients h and K.

(2) Equation 3 shows that the velocity of reheating, $\frac{d\theta}{dt}$, all other things being equal, is proportional to the coefficient $\frac{K}{c\delta}$, where c indicates a calorific capacity far greater than the real specific heat, since it must be great enough to insure the disintegration of the mass. It is important to note that this temperature is not necessarily the ordinary melting point.

M. Charpy has shown in the case of brass alloys this progressive loss of resistance with the temperature much below the ordinary melting point; and aluminum also exhibits characteristics of this order, which probably contribute to its remarkable susceptibility to erosion.

Hence it seems that a study of the amount of erosion, when compared to the known melting points of metals, might lead us to anticipate by anomalies in their classification, certain unique phenomena in the preparatory disintegration of which little is now known.

INCREASE OF EROSION IN THE CASE OF AN UNLIMITED WALL.

I owe to the kind assistance of M. Liouville the solution of a particular case of the problem which is simpler than the one which has just been discussed.

It is the case of an unlimited wall whose surface would be eroded by a current of gas of high temperature.

The conditions at the surface remain the same, but the differential equation 3 becomes—

$$K\frac{d^2\theta}{dr^2} = c\delta\frac{d\theta}{dt},$$

making use of the same notation as before and designating by r the distance eroded measured normal to the surface.

Taking $\frac{K}{c\delta} = A$, an integral of equation 3 is given in exponential form by the expression,

$$\theta = ce^{\left(\frac{m^2t}{A} - mr\right)}, \tag{4}$$

(where m and c are arbitrary constants), which satisfies the conditions at the surface, (2) $\theta = 0$, since $r = \infty$.

Condition (1),

$$-\left(\frac{d\theta}{dr}\right)_{\theta_d} = \frac{h}{K}(\theta_e - \theta_d),$$

determines m.

By differentiating (4) we have

$$-\left(\frac{d\theta}{dr}\right)_{\theta_d} = + m\theta_d = \frac{h}{K}(\theta_e - \theta_d),$$

where

$$m = \frac{h}{K} \cdot \frac{\theta_e - \theta_d}{\theta_d};$$

c is determined by taking as the origin of the time the moment when the surface of the wall with the initial abscissa r_o has reached the temperature of disintegration θ_d;

$$\theta_d = ce^{-mr_o}, \text{ where } \theta = \theta_o e^{\frac{m^2t}{A} - m(r - r_o)}.$$

The velocity of transfer of a certain temperature θ is given by the equation

$$\frac{m^2t}{A} - mr = \text{a constant.}$$

$$\frac{mdt}{A} - dr = 0,$$

where

$$\frac{dr}{dt} = \frac{m}{A} = \frac{h}{AK} \cdot \frac{\theta_e - \theta_d}{\theta_d} = \frac{h}{c\delta} \cdot \frac{\theta_e - \theta_d}{\theta_d}.$$

Hence it is seen that the velocity of transfer of any certain temperature θ is constant.

The admitted hypotheses, therefore, imply the transfer of a calorific wave with the velocity of the erosion.

The outline of this wave would be

$$\theta = \theta_d e^{-m(r-r_0)}.$$

This outline agrees with the surface of erosion following a curve whose tangent deduced above—

$$\left(\frac{d\theta}{dr}\right)_{\theta_d} = m\,\theta_d = \frac{h}{K}(\theta_e - \theta_d)$$

has a considerable value. In fact, the temperature of combustion of explosives generally exceeds by several thousand degrees the melting point of metals.

It is also seen in the case of an unlimited wall or plane that the depth of surface eroded by any point reaching any certain temperature which is capable of changing the metal is constant and independent of the amount of erosion.

The depth of the eroded area will always be constant.

But in passing from the case of a plane to that of a cylinder, in which the analysis is much more complicated, it is easily seen that the rate of decrease of temperature normal to the axis is more rapid for a very small cylindrical hole than for one with a large radius where the action is similar to that on a plane.

These conclusions tend to explain the different value obtained by M. Osmond in the depth of the tempering along the holes of large and of small diameter obtained by the action of the same explosive on the same metal.

CHEMICAL CHARACTERISTICS ACCOMPANYING EROSION.

We have noted the surface tempering constantly observed on the metallic walls which have been submitted to the action of the ordinary explosives.

It has also been observed, through experiments in the lab-

oratory, that through use the internal surfaces of plugs and bombs undergo a veritable cementation which enables them to acquire a temper of a hardness far greater than that which could originally have been given them. This is due to the surface deposits of carbon, which indicates a rather notable dissociation of the products of combustion.

M. Osmond, to whom I am already indebted for a large number of specimens of iron and nickel alloys of notable physical characteristics, has kindly made a micrographic study of the structure of some of the eroded specimens.

His observations are contained in the following memorandum which I have copied literally, with the permission of the author.

MEMORANDUM ON THE SURFACE TEMPERING OF STEEL WHEN SUBMITTED TO THE ACTION OF EXPLOSIVES.

Daubreé, in the course of his experiments on the synthesis of meteorites, was led to place thin sheets of steel in a closed chamber with a powder charge ignited by an electric current. The steel was recovered partly melted; the part which had not been fused had changed from a ductile to an elastic and almost brittle steel, as if it had received a high temper.

In another experiment in which the fusion had been complete, a section of the small ingot, when polished and treated by a weak acid, showed that the structure of the metal was not uniform. The part near the surface stood out very clearly, assuming a different color and following a narrow strip around the edge.

"This difference," says the author, "may be caused by a species of tempering following a very rapid cooling of the surface; however, a chemical action from the inside of the molten mass might contribute to this result."

Unmelted fragments of steel subjected to the action of dynamite gas have shown similar characteristics. (Etudes synthétiques de geologie experimentale, Paris, 1879, pp. 624 à 649.)

Recently I have had occasion to verify these observations of M. Daubreé. M. Vieille sent me five steel cylinders originally pierced by a small axial hole, which had been subjected to the erosive effect of escaping gas of different powders. In examining the microstructure of these cylinders after cutting them in two along the axis, I have immediately noticed the whole length of the corroded walls of the axial hole, a thin strip of highly tempered metal.

The heating and chilling of these strips having taken place under peculiar conditions of pressure and rapidity, it was most interesting to make a careful study of their characteristics.

The five samples will be designated by A, B, C, D and E. A was of mild steel. The other four were of the kind of steel ordinarily used in gun construction. A and B had been simply forged or rolled. C, D and E had been subjected to a retreatment after forging,—probably oil tempering followed by annealing or some similar process.

Sample A.—The strip of tempered metal stood out very clearly after polishing with emery cloth. After a final polish with paste, the difference disappears. The polishing in bas-relief (Fig. 3, 200 diameters) makes the hard spots stand out, the photograph showing them as dark shadows on a clear background, but in the tempered strip, of which the width is about .25 millimeters, these hard grains stand out more clearly and have a finer polish; their area increases in approaching the corroded edge, while at the edge itself they tend to run together. A treatment with iodide colors the hard grains of the principal mass a dark brown (See Fig. 4). These grains are perlite merging more or less into sorbite. In the tempered strip the grains of perlite are abruptly replaced by grains equivalent to martensite (or hardenite, if, as has been suggested by M. LeChatelier, we retain the word which was originally used by professor Howe to indicate saturated martensite). By the treatment with iodide the hardenite is but little colored; towards the surface it enlarges and is encircled by brown, as shown in Fig. 5.

This brown envelope, by its position around the edge of

Fig. 1.—Erosion at Beginning of Rifling.
274 Mm. Gun. Number of shots, 300. Various Powders.

Fig. 2.—Erosion at Beginning of Rifling.
138 Mm. Gun. Number of shots, 1,145. Pure Nitrocellulose Powder.

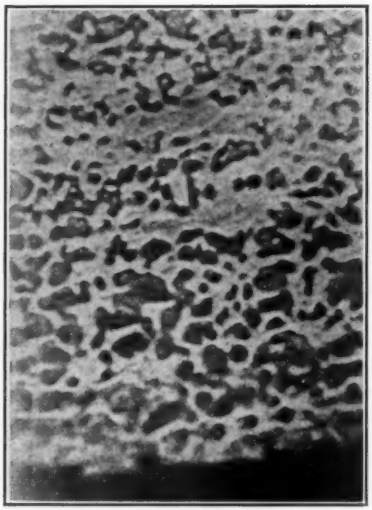


Fig. 3.—Sample A.

MILD STEEL. EROSION BY A POWDER WITH NITROGLYCERINE BASE 50 PER 100.

Charge, 69.17 gr.

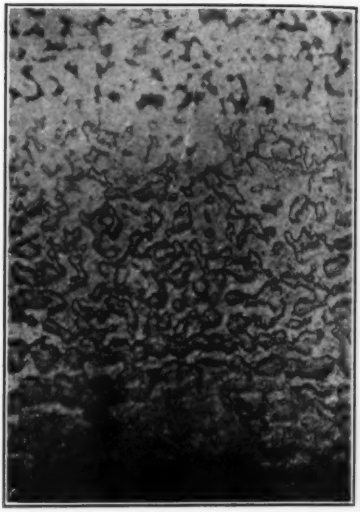


Fig. 4.—Sample A.

TREATED WITH IODIDE.

(200 diameters.)

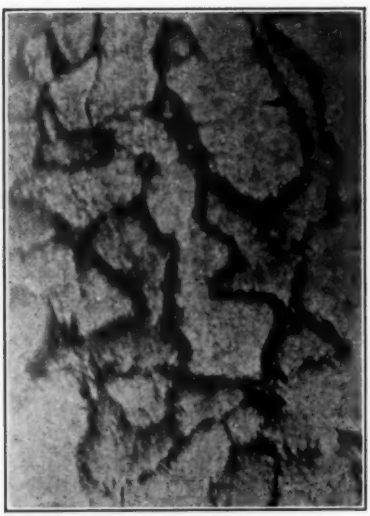


Fig. 5.—Sample A.

TREATED WITH IODIDE.

(1,000 diameters.)

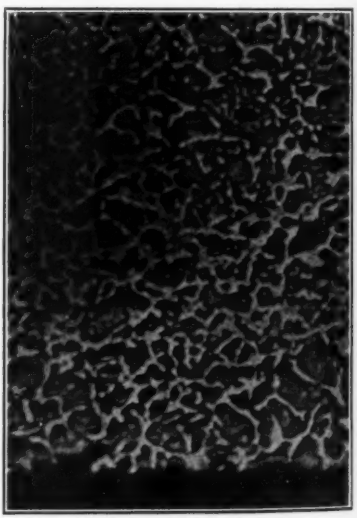


Fig. 6.—Sample B.

GUN STEEL. EROSION BY PURE NITROCELLULOSE POWDER.

Charge, 3.45 gr. (200 diameters.)

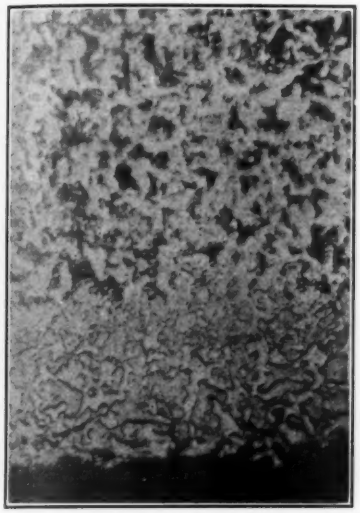


Fig. 7.—Sample B.

TREATED WITH IODIDE.

(200 diameters.)

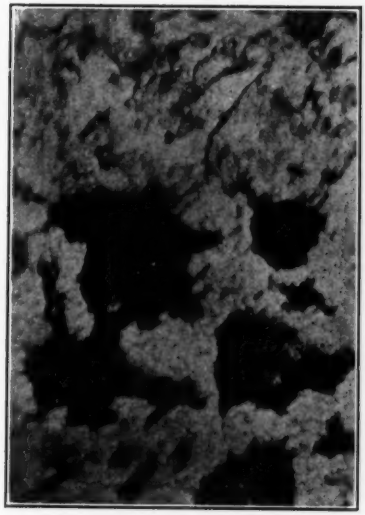


Fig. 8.—Sample B.

TREATED WITH IODIDE.

(1,000 diameters.)

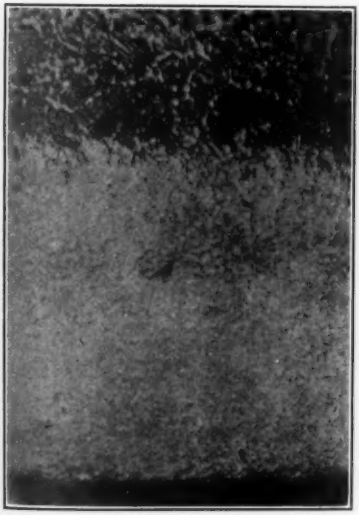


Fig. 9.—Sample C.

GUN STEEL. PURE NITROCELLULOSE POWDER.

TREATED WITH IODIDE.

Charge, 69 gr. (200 diameters.)

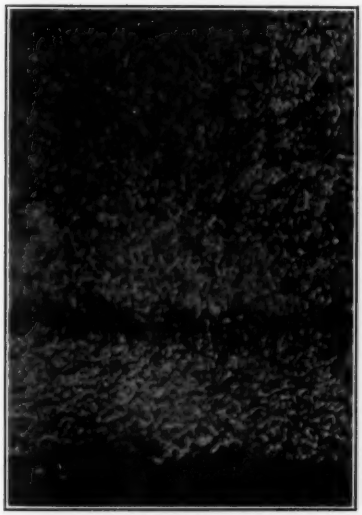


Fig. 10.—Sample D.

GUN STEEL. NITROGUANIDINE POWDER.

TREATED WITH IODIDE.

Charge, 3.50 gr. (200 diameters.)

the hardenite, corresponds to troostite, but with the peculiarity that it presents a trace of needles, hardly visible in the photograph, which resemble the structure of martensite. We have then a secondary form of transition between martensite and troostite which has never been described, and this form of transition corresponds to the beginning of the diffusion of carbon from the hardenite to the ferrite.

In approaching the edge the hardenite which formed the heart of the bordered grains becomes milder, on account of the increasing diffusion of the carbon, and by renewing the treatment with iodide the brown color invades the whole grain without preventing, however, the characteristic form of the martensite from appearing clearly. These forms are not found except at the extreme edge of the corroded surface, all remaining relatively fine and confused; the distribution of the carbon has not yet become homogenous, as may be seen from an examination of Fig. 3. The clear network which continues throughout to separate the dark grains represents the soft martensite, which passes gradually to the hard martensite used in cores.

These remarks are easily analyzed.

The gas escaping at a high temperature and at great pressure, heats very suddenly the walls of the axial hole, the temperature decreasing rapidly towards the interior of the metal. Then a chilling no less abrupt takes place, either from the mass of steel that has not had time to become heated, or through the expansion of the gas during the latter part of its flow.

The line which separates the tempered from the untempered mass represents the depth to which the heat penetrated at the beginning of cooling, the point Ac 3, 2, 1, of the perlite. The transformations of the carbon and the iron have taken place, but on this limiting line the chilling has followed the heating so rapidly that the diffusion of carbon has not had time to begin, and the perlite has given place to the hardenite. Approaching the edge, layers of metal are found which underwent a progressive increase of heat in a propor-

tional increase of time, all, however, very short. The carbon has begun to separate itself from the grains of hardenite, and is diffused more and more, but without being able to attain a homogeneous distribution in the last layer, where the original network of iron still remains represented by soft martensite. The layers which underwent the greatest heat have been carried away by the gas.

Sample B gave similar results. Fig. 6 (200 diameters) shows the appearance in bas-relief after polishing; there is less ferrite than in A on account of the higher percentage of carbon. The tempered strip is only $\frac{1}{8}$ mm. in thickness, and the layer of pure martensite is not represented; the network of ferrite, much rarified, extends to the eroded surface. Fig. 7 shows the same metal lightly treated with iodide. The perlite appears dark, the hardenite and the martensite are lightly colored, while the ferrite remains white, or slightly granulated. In Fig. 8 (1,000 diameters, treated with iodide) a mixed grain is shown where the sorbite changes abruptly to hardenite; immediately underneath, a grain shows the sheets of perlite, but so fine that they can hardly be reproduced by photography.

Samples C, D and E.—The mass of these samples shows the normal structure of gun steel, oil-tempered and annealed; that is, ramified strips of ferrite on a poorly-defined background, which represent the forms of transition between perlite and martensite, or, more exactly, between sorbite and troostite.

As in samples A and B, the surface of erosion is bordered by a sheet of tempered metal, but on account of the previous treatment of the metal, the distribution of carbon was comparatively uniform before the escape of gas, and the greater part of the tempered strip is here formed of martensite.

Fig. 9 represents sample C; the tempered band is .25 millimeter thick. An extended treatment with iodide leaves the martensite clear while the larger unchanged mass is highly colored, with the exception of the strips of ferrite; the latter penetrate nearly one-third of the width of the belt of marten-

site. At the limit of the tempering towards the interior a very dark band is noticed. This band represents the region where the temperature is raised by the passage of the gas to a degree exactly under the critical point. Tempering is consequently not possible, but the effect of the original treatment of the steel has been effaced to a certain extent, and the metal resembles sorbite.

The band of sorbite is also clearly seen in Fig. 10, sample D (200 diameter, treated with iodide). The same photograph could also be considered as sample E. The thickness of temper is but $\frac{1}{10}$ of a millimeter, and as the initial ferrite was rare, it is formed almost exclusively of martensite.

THE ELEKTRA STEAM TURBINE.

By Dr. H. Meuth.

Extract, translated from "Zeitschrift Vereins Deutscher Ingenieure,"* by Philip Pistor, M. E.

A turbine of the impulse type is that invented by Civil Engineer O. Kolb, of Karlsruhe, which is known by its name of the Elektra Turbine. It is being manufactured in Germany by the Company for Electrical Industry, at Karlsruhe, and also in the navy yards of the imperial Navy; furthermore in Austria by the Skoda Works, at Pilsen; in Belgium by the machine shops, forge and foundry stock company, in Moustier sur Sambre; in France by the Compagnie General Electrique, at Nancy; in Spain by the S. A. Construcciones Mecanicas Electricas, in Gerona, and in Italy by the Officine Elettro Meccaniche, in Genoa.

This purely pressure turbine utilizes the energy of expansion of the steam, as does the Curtis turbine, in this manner: the steam tension is first totally transformed in a nozzle into velocity, and the kinetic energy of the steam jet obtained is then transformed into mechanical work in a rotor step by step. The Elektra turbine differs from the Curtis turbine, which is characterized by several running blade rings and intermediate fixed blade rings; and apart from its construction as a radial-flow turbine, in that the speed step drops and, with the same the reduction of the rotor speed, is accomplished by a very simple mechanical device.

Figures 1 and 2 show sections of the Elektra turbine. The steam enters at *a* into a channelway which surrounds the turbine housing and reaches the nozzles *p*, which lie diametrically opposite to each other. The steam, which is expanded down to

*See Zeitschr. V. D. I., p. 132, Feb. 1, 1908. Compare also 1904, p. 1556, and 1906, p. 1415.

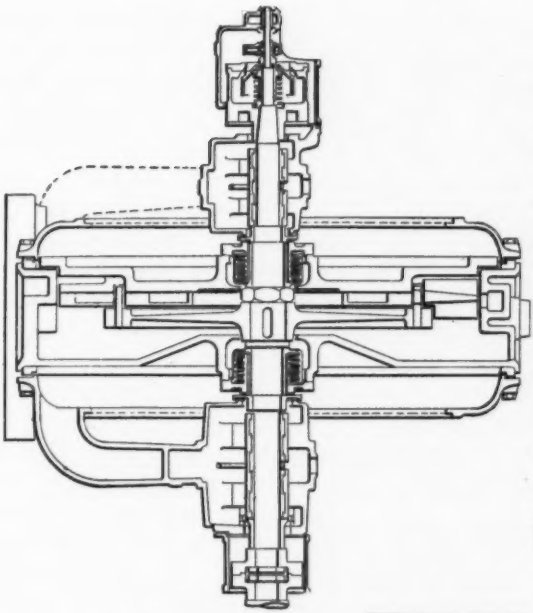


Fig. 2. Compound Turbine With Two Casings.

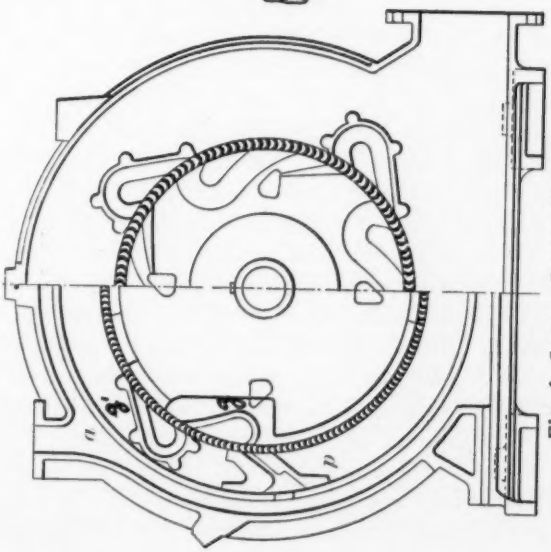


Fig. 1. Single Casing Turbine.

the counteracting pressure of the atmosphere or to that of the condenser, impinges on the axial blades of the rotor (i. e., placed parallel with the axis), and on its exit from the rotor is received by a rectangular channelway g, which at first takes the same direction as the issuing steam jet; in its further course, however, it is so curved that the passing steam impinges a second time on the bladed rotor at an angle determined for an entrance without shock. The same action is repeated in a second and third diverting channelway g^1 and g^2 respectively; the width of the channelway in the direction of its deflection is increasingly widened to correspond to the decreasing steam velocity.

From the last channelway the steam passes through the wheel into the turbine housing and through the nozzle into the open air or into the condenser. The fourfold impingement on the rotor makes it possible to obtain an economical utilization of a high steam velocity with a relatively low rotor speed (about 100 meters at the circumference), which makes no extraordinary demands on material or construction. The reduction of the rotor speed is accomplished, as is seen, in the most simple manner. In addition to this extraordinarily simple construction come now the advantages of the turbine as a free jet turbine: the whole tension of the steam is transformed into velocity before reaching the rotor, that is, away from the only movable part. Neither the high pressure, nor with the manifold use of hot steam does the high temperature of the entering steam, to which only the nozzle as well as the seat and body of the stop and regulating valves are exposed, enter the turbine proper, whereas rotor, diverting channelways and housing walls only come in contact with low-pressure cooled steam. The turbine can fully utilize the advantages of high superheat without risk; it has been demonstrated that thereby not only an improvement of the thermal efficiency is gained in consequence of the higher current energy per kg. of steam, but also a lessening of rotor friction and blade wear, particularly if the superheating is carried so far that the steam at the end of nozzle is still superheated. The turbine is ready

for work without further preparation and needs no time robbing precautionary measures at starting, in case a reliable working water separator is incorporated in the steam main. The equality of pressure at the passage of the steam through the rotor makes a packing of the slit at the circumference of the wheel needless. The slit, on the contrary, may be made so large that a rubbing of the wheel, even after a deformation of the housing from heat and on account of minor defects in erection, is surely prevented. Axial pressure on the shaft, which requires a complicated compensation in high-pressure turbines, does not occur in balanced-pressure turbines if the slight effect of the blade friction is neglected. With radial action, as here with the Elektra turbine, an axial thrust does not occur anyway. In consequence of the symmetrical arrangement of the nozzles only a purely torsional moment is exerted on the shaft, and no free force that could exert a one-sided influence over it.

The advantages of the balanced-pressure turbine with lowering speed steps are, however, connected with the disadvantage of greater frictional resistance in the blades and deflecting channelways on account of the greater relative steam velocity, particularly in the first step. The friction has as a consequence that the steam velocity decreases far more rapidly than what would correspond to the step-like surrender of work to the rotor, and that with a small number of steps a sufficiently low terminal velocity is reached; usually four are not exceeded. Up to the present a wear of the blades has only been markedly noticed in wet steam.

For the larger powers (over 50 H.P.) the Elektra turbines, as a rule, are constructed with two pressure steps, each with three velocity steps. The housing of such a compound turbine is divided by a partition into two chambers; each forms a separate turbine, and works exactly like the simple turbine first described. With a normal load the steam expands in the nozzles of the front turbine from initial pressure down to about 1 atmosphere absolute, then flows through openings in the partition into the nozzles of the second turbine, and ex-

pands in these down to condenser pressure. The velocities of flow of the steam and the consequent frictional losses are noticeably less with this division of the pressure. With the same rotor velocity as in the simple turbine the compound turbine attains a very favorable thermal efficiency without losing much of simplicity.

CONSTRUCTIONAL ELEMENTS OF THE ELEKTRA TURBINE.

Figure 1 shows the short axial length of the turbine. With so short a distance between bearings and the light weight of wheel and shaft, simple oil-ring bearings will suffice, and only with large turbines are they cooled with water. The absence of the pressure pumps for pressure lubrication represents simplicity and safety in running. In the larger turbines the bearings are not carried on the housing cover as shown in sectional Fig. 1, but are supported directly on the bed-plate, whereas the housing is carried concentric with the bearing body by a flange on the same and can expand freely in every direction.

The bronze expanding nozzle is inserted in a cast-iron chamber and bolted fast to the housing wall. The cross section of the nozzle is rectangular and expanded towards the exit. One side wall is made movable and fitted steamtight at the sides; this steel tongue can be adjusted from the outside by means of toothed gearing. In general, however, the nozzle remains in its normal position for all loads under full load; the nozzle is widened only for the purpose of temporary overload of the turbine. The nozzles of the second pressure step in compound turbines are not adjustable.

The rotor consists of a steel plate, on the rim of which the blades are laid closely together and are secured by a heavy clamping ring. In revolving a moment is generated on account of the centrifugal force of the axial overhung blades, which is taken up by a light clamping ring on the outer end of the blades. This requires a certain relation of its cross-section clamping dimensions to the corresponding dimen-

sions of the main clamping ring. The latter is of such dimensions, and is forced on in such a manner, that, regardless of the arising moment, its clamping strength can balance the strains due to its own centrifugal force; furthermore, the pressure of the blades and that due to the disk which enlarges in revolving, or rather exceed these forces by a safety factor (about 50 kg. per 1 cm. of the circumference), so that the combination will not yield under any conditions. The outer small clamping ring must follow the radial extension of the blades and finally exert a pressure on the rim blades, which balances the pressure caused by the centrifugal moment of the projecting part of the blades. Only in this case will the main clamping ring be free from twisting strains.

The blades are cut to specified length from extruded Ruebel-bronze. The front and back blade surfaces are milled on special tools. Steel blades have not stood the wear in saturated steam; rather use bronze blades which have shown no wear in superheated steam.

A notable part of the Elektra turbine are the water packings of the shaft. It may be seen from Fig. how a row of thin plate disks secured in heavier rings are arranged at intervals of 1.5 to 2 mm. side by side; alternately a disk is secured to the housing and one revolves with the shaft. Sufficient play is also left radially, so that in case of a relative displacement of shaft and housing, in consequence of uneven heating, the disks will not touch. Water is introduced into the intermediate spaces between the plates from the outside, which is carried along by the friction of the revolving disks and thrown outwards. In connection with the arising contraction, the centrifugal pressure of the water in the separate chambers, if their number and the speed of revolution is sufficient, will balance an inner or outer surplus of pressure, which, however, at most will be 1 atmosphere. A safety valve to provide for too high a pressure in the housing is unnecessary, as the steam can blow off through the clearance spaces. With this packing metallic parts come nowhere in contact, therefore all friction, wear and lubrication are eliminated. The packing water,

which should be as clear as possible to avoid stopping up, may be taken from a water main. The water supply for the front and back packing is installed separately and in such a manner that it only passes off drop by drop. The consumption is then very small, about 30 to 40 liters per hour. The power loss in consequence of the friction of the plates in the water is also very small. With a turbine of 250 H.P. the work of running idly of turbine and dynamo at 3,000 revolutions and 60 per cent. vacuum was 9.66 kw. with packing water, without packing water it was 7.79 kw. The difference of 1.87 kw. therefore gives a loss of energy for the packing of about 1 per cent. of the whole duty. With smaller turbines the frictional work is a higher percentage, owing to the relatively but little smaller lever of the friction, and amounts to about 8 per cent. in a 20-H.P. machine.

In the small single-step turbine of 8 H.P. the packings are fed with steam. In compound turbines the chambers of the first and second pressure steps must be packed to avoid blowing through. This is done with grooves in the shaft, which at this point is surrounded by a babbitt box.

The shaft is axially secured by two collars at the bearing on the coupling end. The bearing shell has a short, flat screw thread on its end. With the aid of the nut held fast by the bearing body the bearing shell, and at the same time the shaft and all parts secured to the same, may be shifted axially relatively to those parts secured to the housing and accurately adjusted. The dimensions of the shaft are fixed so that the vibrations of the revolving parts proper are several hundred above the normal number of revolutions. The turbine shaft is coupled to the dynamo shaft or to a jack shaft by means of a set of leather disks between the two coupling flanges. The arrangement allows of simple construction and ready accessibility to all parts of the turbine. After removing the front bearing shell and cover the first step rotor can be drawn from the shaft; after removal of the nozzle chambers and the middle diaphragm the second rotor is also exposed.

Regulation is done by throttling the supply steam, as with

most of the present built turbines. Formerly attempts were made to control the steam by the more correct method of changing the nozzle openings, but abandoned on account of the difficulty of bringing the adjustable nozzle in any simple manner under the positive control of the regulator. In the present method of throttle regulation by means of a perfectly balanced double-seated valve a spring governor actuates the throttle directly through a sleeve even with the largest units. The principle of the regulator is the same as in the de Laval turbine. The regulator (or governor) spindle is an extension of the turbine shaft, see Fig. 2. With the high number of revolutions of this shaft the governor becomes very small, but has nevertheless a considerable power of motion and great energy, which renders it capable of controlling the unavoidable strains transferred through the regulator and connections; such are caused by the passage of the steam through the balanced valve.

If carefully constructed the governor is very sensitive. By the size and suspension of the rotating weights the variations in speed between full load and no load may be brought to 1 or 2 per cent. of the normal revolutions.

In the housing of the governor there is further a revolving spring-loaded weight which strikes a bent lever when the set number of revolutions is exceeded, which is about 5 per cent. above the normal, and by means of linkage releases a lever on the throttle. This holds at its end the divided nut of the valve spindle securely together, like a crab claw. On its release the spindle is freed from its trapezoidal-shape thread and the valve immediately closes under the action of a spring. This self-closing device, however, only comes into operation in case the normal regulation should fail on account of an accidental disarrangement.

In order to maintain the operation of the turbine with its normal load in case of any accident to the condensing plant, special nozzles have been provided for working with open-air exhaust. They are so proportioned that in combination with them the whole duty of the turbines is realized (in compound

turbines in connection with nozzles for driving the condenser). The change from one to the other manner of running is accomplished without interruption. In compound turbines the work of the second wheel is sacrificed, which anyway is inconsiderable, as the nozzles and diverting channelways of the second-pressure stage are not suited to the lowering pressure arising with an open-air exhaust. The second rotor is therefore allowed to run idly, and the steam is allowed to pass directly into the second chamber through openings otherwise closed. The pressure of the exhaust chamber is then developed in the chamber of the first step, and the gain by lessening of rotor friction in the rarer steam in most cases overbalances the small quantity of work of the second rotor, had the steam been conducted through the nozzles of the second step.

STEAM CONSUMPTION.

A single-step Elektra turbine of 20-H.P. normal capacity gave the results in Table I, according to the measurements of Professor Stodola on June 12, 1906.

The results from a two-step turbine of 100-H.P. normal capacity with 11 atmospheres initial pressure (absolute) before entering the nozzles, 250 degrees steam temperature, 92 per cent. vacuum and 3,000 r.p.m., are graphically depicted in Fig. 10.

Experiments with a compound turbine of 300 H.P. coupled to a polyphase generator of 2,000 volts gave the results in Table II and Fig. 11.

A turbine of the same size and build, working, however, with saturated steam of 9 atmospheres absolute initial tension, was investigated by Professor Hubert Luettich on May 27, 1907, and gave the tabulated results of Table III and Fig. 11.

The graphical representation of the hourly steam consumption shows a straight-line increase with the electrical output, as has been found with other turbines. The hourly steam consumption for various part loads may therefore be determined from the simple equation $Dhr^{*}=a\,L+b$, in which a is

* Dhr = steam per hour.

a constant that in one and the same machine depends on initial pressure, the superheat and the vacuum, whereas b is the steam consumption of the idle running machine; L is the load.

The steam consumption results given in Table IV were attained by an Elektra compound turbine of 450 H.P. according to measurements taken by Professor Brauer, of Karlsruhe.

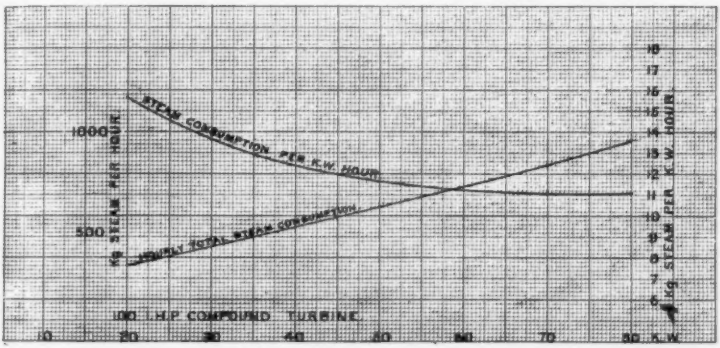


Fig. 10.

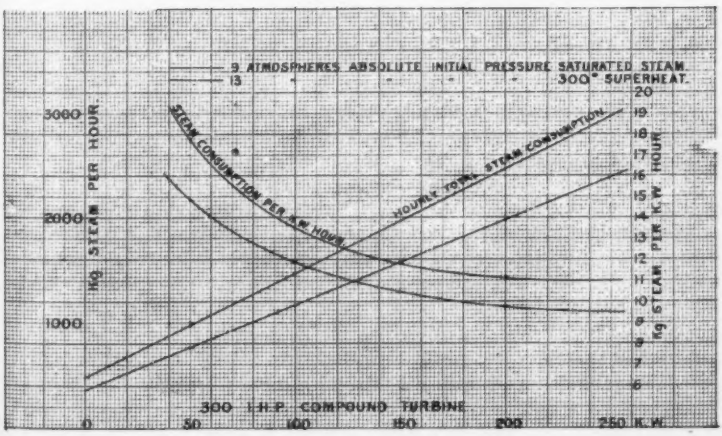

9 ATMOSPHERES ABSOLUTE INITIAL PRESSURE SATURATED STEAM.
13 " " " " 300° SUPERHEAT.
STEAM CONSUMPTION PER K.W. HOUR.
HOURLY TOTAL STEAM CONSUMPTION.
Kg. STEAM PER HOUR.
Kg. STEAM PER K.W. HOUR.
300 H.P. COMPOUND TURBINE
K.W.

Fig. 11.

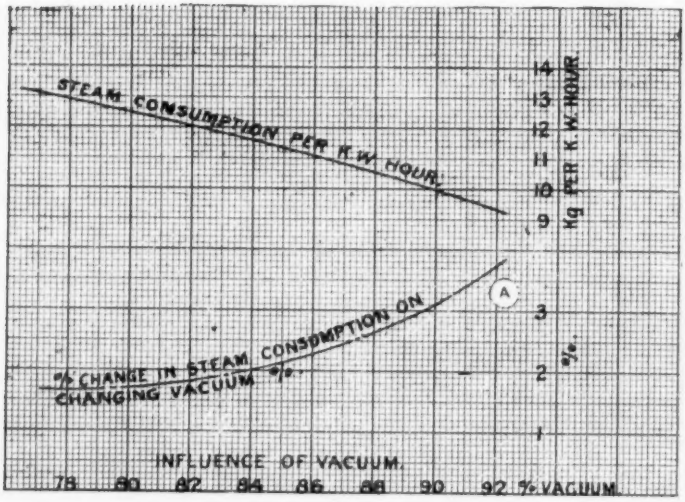


Fig. 12.

The condensing plant gave no higher vacuum than 88 to 89 per cent. In order to determine the action of the turbine with a guarantee vacuum of 90 per cent. for which it had been built, three experiments were made with 78, 83 and 88 per cent. vacuum; steam pressure and temperature, nozzle setting and revolutions were kept unchanged throughout. The result is graphically represented in Fig. 12. The influence of the vacuum is plainly shown by the course of the plotted curve A, which represents the decrease in percentage of the steam consumption for 1 per cent. increase in the vacuum. At 90 per cent. vacuum, for instance, the steam consumption is lessened by about 3 per cent. if the vacuum increases 1 per cent. The theoretical gain comes to about one half of this value, as the friction in the turbine decreases with rising vacuum.

The steam consumption figures in Tables I to IV have been obtained by measuring the condensed water; the added values of the thermal efficiency relate to the condition of the steam before entering the nozzle.

TABLE I.

Load.	Steam before the nozzle.		Vacuum, at. ab.	Temp. exh. ° C.	Revs. per min.	Performance.		Steam consumption.		Thermal efficiency, per cent.
	Press. at. ab.	Temp. ° C.				Kw.	H.P.	kg. hr.	Per H.P. per hr.	
	12.4	256	0.105	106	3,925	16.5	25.8	310	12.0	28.0
	12.4	245	0.087	88	3,940	12.3	19.7	248	12.6	26.0
	7.6	204	0.105	86	3,900	4.9	9.3	147	15.8	25.3

TABLE II.

Load.	Steam before the nozzle.		Steam in intermed. chamber.		Exhaust.		Revolutions per minute.	Performance.		Steam consumption.		Thermal eff., per cent.
	Press. at. ab.	Temp. ° C.	Press. at. ab.	Temp. ° C.	Press. at. ab.	Temp. ° C.		Kw. hr.	Per H.P. per hr.	Kg. hr.	Pr. H.P. per hr.	
	13.7	290	1.15	111	0.08	43	3,000	202.0	308.0	1,968	6.45	48.5
	10.9	272	0.89	103	0.08	42	3,000	150.5	235.0	1,580	6.75	48.7
	8.5	260	0.66	79	0.07	42	3,000	106.4	172.0	1,269	7.38	46.0
	4.9	245	0.35	85	0.07	34	3,000	50.0	85.0	741	8.77	43.5

TABLE III.

Load.	Steam before the nozzle.		Steam in inter. chamber, at. ab.	Exhaust.		Revolutions per minute.	Performance.		Steam consumption.			Thermal eff., per cent.
	Press. at. ab.	Temp. ° C.		Press. at. ab.	Temp. ° C.		Kw. hr.	Per H.P. per hr.	Kg. hr.	Per H.P. per hr.	Per H.P. calc. for 90 per ct. vacuum.	
	9.5	174	1.46	0.12	46	3,005	255.0	385	3,029	7.85	...	51.0
	8.6	170	1.15	0.10	42	2,990	204.6	312	2,428	7.80	...	50.5
	6.9	167	0.90	0.08	38	3,000	152.0	237	1,892	7.95	...	50.0
	5.4	160	0.70	0.07	36	3,000	108.2	174	1,479	8.50	...	48.5
0	1.0	114	0.07	0.053	38	3,020	...	...	314	...	...	...

TABLE IV.

	13.5	298	1.05	0.118	60	2,010	310.8	458	3,129	6.82	6.45	51.5
	13.3	286	0.83	0.113	55	2,000	247.0	373	2,567	6.87	6.60	51.0
	11.3	294	0.75	0.106	61	1,980	203.3	313	2,248	7.18	7.06	48.1
	9.1	290	0.60	0.107	65	1,975	147.2	224	1,835	8.25	8.11	43.6
	7.2	272	0.45	0.115	64	1,990	103.3	165	1,525	9.15	8.85	42.0

FIELDS OF USE FOR KOLB'S ELEKTRA TURBINE.

First we will have to consider direct driving of dynamos for direct and alternating current. At first the building of generators suitable to the high number of revolutions of the steam turbine gave much trouble in dynamo construction. It was necessary to break away entirely from normal conditions of construction.

A further field for the turbine is the direct running of centrifugal pumps, centrifugal machines, blowers and rotary compressers. The normal number of revolutions of the steam turbine is sufficiently high for these machines. Blowers, especially as they are used in great numbers aboard ship for ventilating and for forced draft, make relatively a small number of revolutions, about 1,000 to 1,200 per minute; at higher speeds the flywheel action becomes too unfavorable. Here the turbine designer has the problem to construct a slow-running turbine for relatively small duties of 30 to 50 H.P. Such a turbine of 50 H.P. at 1,200 r.p.m. will have a single-bladed wheel of 800 mm. diameter, on which the steam impinges six times. The machine is built for a back pressure of 1.5 atmospheres absolute, and is intended for installation on board ship. For this purpose the Imperial Navy prescribes very stringent requirements, greatest simplicity in design and in operating, as well as the quality of ready response to the varying conditions on board of men of war, on which the demand for air at times varies within wide limits. The turbines must be able to deliver about one-half their normal duty at 700 r.p.m. At the same time they must not materially exceed the space requirements, the weight and the steam consumption of the highly-perfected vertical high-speed engines with their 19 to 20 kg. at normal duty and 80 per cent. vacuum. As regards simplicity, the turbine can hardly be surpassed. The steam is led to the wheel through two nozzles lying opposite to each other, whose chambers are connected by a w.i. pipe. The connection of the overhung turbine casing to the main bearing is so designed that the turbine may be turned 90 degrees, and

thereby connections for steam and exhaust can be more readily adapted to local conditions. The shaft is enclosed in a box with grooves turned in at the point where it projects from the casing; a packing outside further makes it tight to inner and outer higher pressure.

There is no special regulator for changing the duty; the number of revolutions will be changed by hand by throttling the live steam at the main stop in accordance with the greater or lesser demand for air. For accidental heavy increases of the normal number of r.p.m an automatic device of the above mentioned kind is attached. The attention required by the machine is limited to starting and stopping; the bearings have automatic lubrication.

The result of duty tests undertaken under the direction of the higher yard officials, on a 30-H.P. turbine with condensation and with a normal 1,000 r.p.m. was obtained. In order to measure the turbine duty at different r.p.m., the machine was coupled up to a direct-current dynamo whose capacity had been accurately determined for the different revolutions. At 10.5 atmospheres absolute steam pressure and 194 degrees C. steam temperature, 0.23 atmospheres absolute back pressure and 1,020 r.p.m., a steam consumption of 39.16 lbs. per H.P. per hour was recorded, which, with the same passing quantity of steam, but only 590 r.p.m., rose to 62.04 pounds per H.P. per hour. The machine developed in the first case 32.9 H.P., in the latter 20.9 H.P. When the nozzle was closed slightly, there resulted with the same pressure conditions 695 r.p.m., a development of 16.7 H.P., and a steam consumption of 25.4 kg. per H.P. per hour. With wide open nozzles and 10 atmospheres absolute initial pressure a development of 60 H.P. was recorded. The machine is consequently extraordinarily adaptable to its duty.

Slowly running Elektra turbines as motors for centrifugal pumps are particularly advantageous in those cases where the exhaust is utilized. A high vacuum has been shown to give but small gain; the higher energy of the current is only imperfectly utilized with the low circumferential speed of the

wheel, and will be saved again in part in the heat of the exhaust. If the exhaust of the auxiliaries is used, as aboard ship, for feed heating, or, as in operating centrifugal machines in sugar houses, for boiling, then the installation is also very economical, besides its extraordinary simplicity.

The Elektra turbine, in consequence of its peculiar principle of design, may be easily arranged for reversing. The rotor carries overhanging blades on both sides for this purpose. The one blade ring serves for going forward, the other with blades placed in the opposite direction, for going backward. By means of a reverse valve the steam may be led to the nozzles of the one or the other side; one of the blade rings then runs empty. For driving large ships the Elektra turbine is not to be considered. Apart from its capacity limit, about 1,000 H.P., which is determined by the nature of its design, turbines with few rotors have too small a torque moment.

COAL CONSUMPTION OF FRENCH TORPEDO-BOAT DESTROYERS WITH TURBINES, RECIPROCATING ENGINES AND COMBINATION MACHINERY.

By Ernest N. Janson, Member.

The data in the following article were given by Professor Rateau, of Paris, and incorporate results obtained with three destroyers of the French Navy in recent official trials. Professor Rateau spoke of said trials and the results obtained as demonstrating the advantages of the combination system.

To arrive at the best possible type for a certain class of torpedo boats, the French Marine resorted to competitive proposals from the French shipbuilding establishments. Six projects were received, of which three were adopted having reciprocating engines with forced lubrication, the *Huzzard, Spahi* and *Carabinier;* one with Parsons turbines alone, the *Chasseur,* by Normand & Cie., du Havre, figure I; one with a reciprocating engine, and two Rateau turbines, the *Voltigeur,* by de la Brosse & Fouché de Nantes, figure II; and, lastly, one, the *Tirailleur,* with reciprocating engines and turbines, by Bréguet, of the de Laval type.

The dimensions of the boats are: length, 195 feet; breadth, 20.5 feet, and molded depth, 13.5 feet, with a draught forward of 6 feet and aft of 7 feet on 450 tons normal displacement.

DESCRIPTION OF "VOLTIGEUR'S" MACHINERY.

The *Voltigeur* has four Normand boilers, the same as the *Chasseur.* The propelling machinery is composed of one 3-cylinder triple-expansion engine, having forced lubrication, driving the center shaft, and two Rateau turbines, one each on the wing shafts. The engine is of about 2,700 H. P., with cylinders 18.3 inches × 26.4 inches × 40 inches by 21 inches

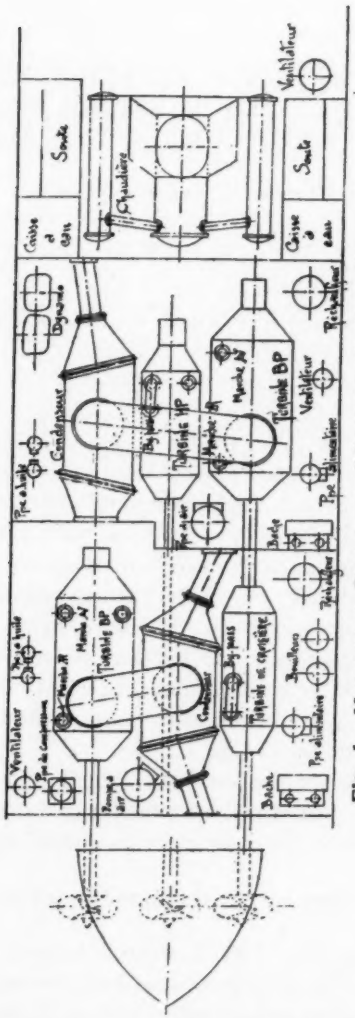


Fig. 1.—Machinery, Destroyer "Chasseur." Plan View.

stroke, and 380 revolutions per minute. It drives an attached air pump and exhausts into its own condenser when used for cruising speeds, between 14 and 21 knots. For higher speeds

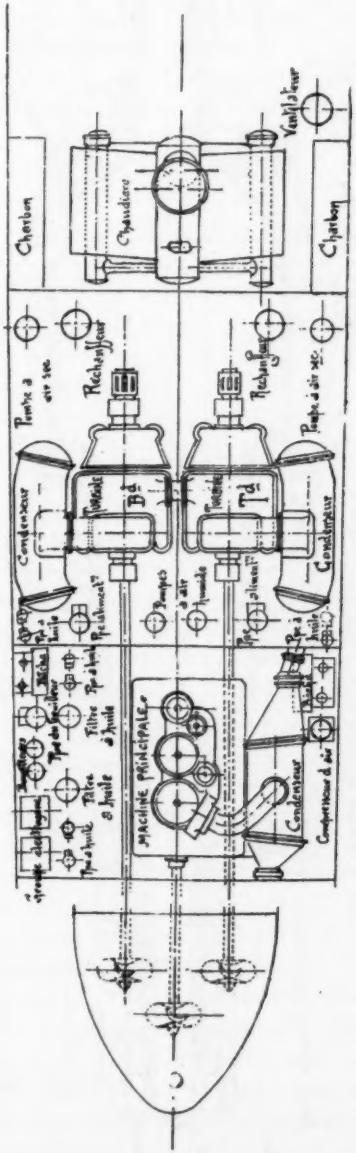


Fig. 2.—Machinery, Destroyer "Voltigeur." Plan View.

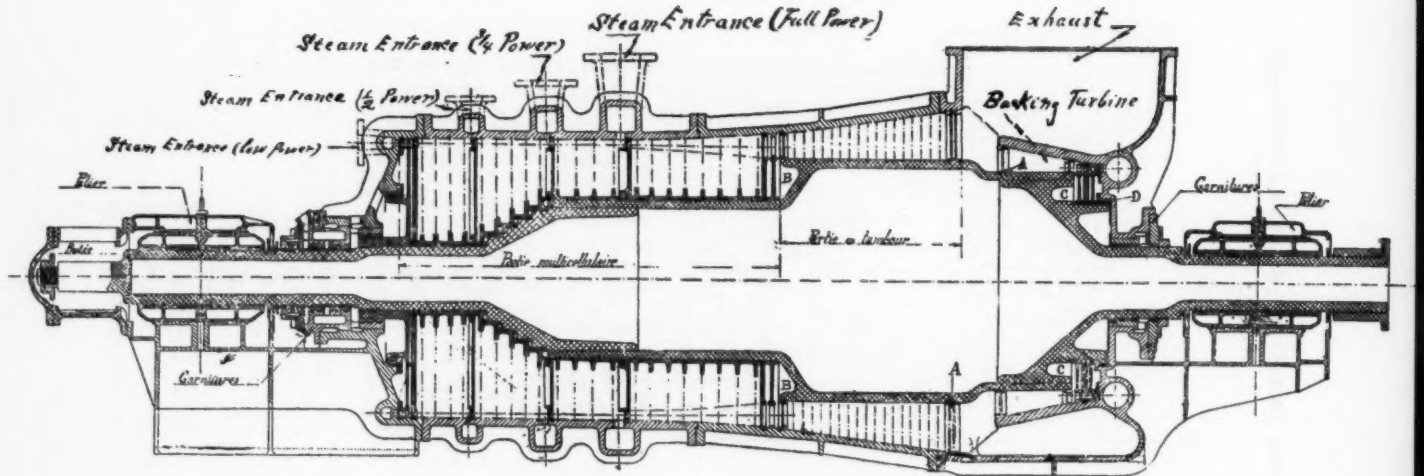


Fig. 3.—Rateau Marine Steam Turbine.

it exhausts directly into the L.P. turbines. All the auxiliary machinery, ship and engine, exhaust in this condenser.

In the engine condenser scoop-injection is used for running ahead while an independent circulating pump is used during stops and as an auxiliary. The cylinders are not steam-jacketed but the bottoms and covers are. There are piston valves on H.P. and I.P. cylinders and flat valves on the L.P. cylinder, all actuated by Stephenson links. Bedplates and bearing pedestals are of cast steel. The two Rateau turbines are placed in the same compartment, each turbine consisting of ahead and backing turbines within the same casing. The high and medium-pressure zones are composed of discs with single vane rows while the vane system in the low-pressure zone is mounted on a drum. On the forward end of each turbine is a thrust bearing. The internal arrangement is such as to counteract the thrust exerted by the screws.

When going ahead, the wing screws turn outboard and the center screw turns in the same direction as the starboard screw, which is right-hand. There are four separate steam inlets to the turbines, one for low speeds, one for half speed, another for three-quarter speed and finally one for maximum speed. Separate service motors operate the throttles to the ahead and backing turbines as well as the valves in the main exhaust pipe, diverting the exhaust steam from the engine to either the engine condenser or to the turbines. The turbine condensers are large enough for maximum powers, and in connection with them are operated two independent air pumps, one wet and one dry pump, making in all four air pumps, besides the attached one on the engine. Separate circulating pumps are run on the turbine condensers for the circulating water. The air pumps are of the Westinghouse Leblanc rotative type.

Attention was drawn to the fact that even for speeds much below cruising speeds of 21 knots some advantage in economy was obtained by combining the powers of engine and turbines. The amount of the energy retained by the steam after its release from the engine may be judged from the hatched area in figure 4, the diagram of combined indicator cards.

Theoretically it corresponds to a force equal to the mean pressure multiplied by a piston area in size such as to allow this pressure being obtained. This, however, is not possible with the engine, but is accomplished by connection to the turbines.

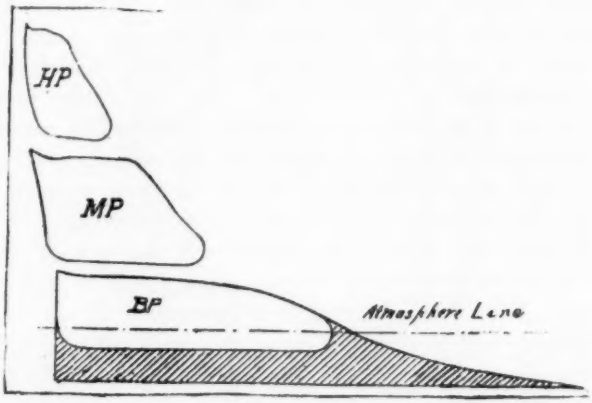


Fig. 4.—DIAGRAM OF TRIPLE-EXPANSION ENGINE EXHAUSTING TO TURBINES.

The hatched part represents the supplementary work saved by the turbines.

The turbine screws are in the same transverse plane, the center screw being abaft of them. Each turbine is calculated for 3,000 H.P. with 850 revolutions per minute. The backing speed is from 30 to 40 per cent. of the ahead speed with the turbines alone.

TRIALS.

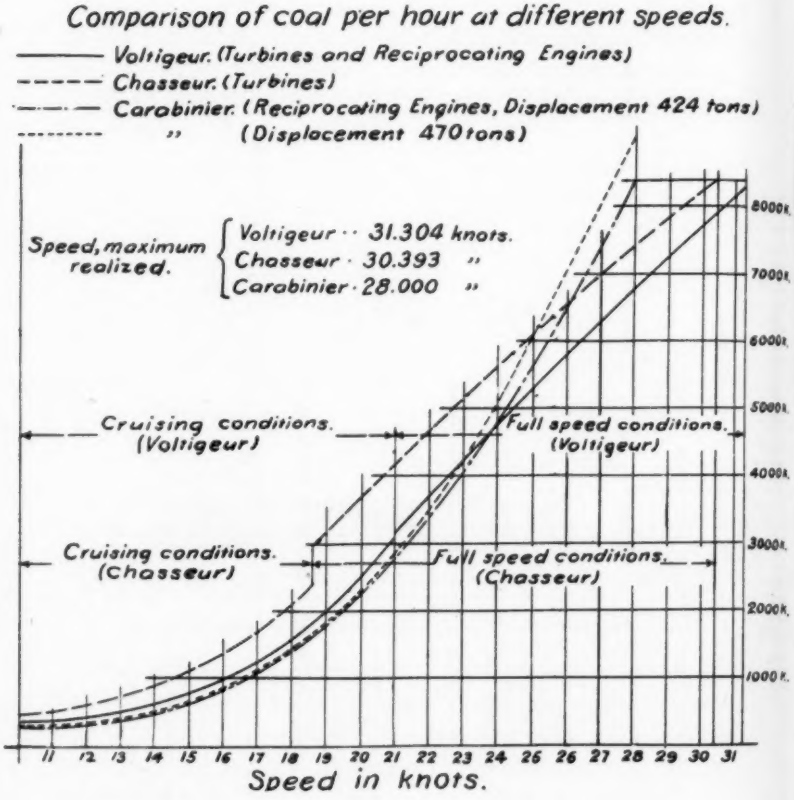
The torpedo boats *Chasseur* and *Voltigeur* were run between Cherbourg and Lorient under as nearly as possible the same conditions. The principal trial results of these two boats, as well as those obtained with the torpedo boat *Carabinier*, although on somewhat smaller displacement and under conditions incidental to her having reciprocating engines, are given in the annexed table.

On the curves speeds in knots are plotted on the abscissae and coal consumption per hour on the ordinates.

COAL CONSUMPTION AT DIFFERENT SPEEDS.

One-Hour Trial, at Maximum Speed.

.....	*Chasseur.*	*Voltigeur.*	*Carabinier.*
Mean speed realized during one hour, knots.....	30.393	31.304	27.7
Coal burned, total per hour, tons.....	8.48	8.28	8.4
per square foot G. S. per hour, pounds	81.0	77.7	80.9
mile covered during trial, pounds.....	619.0	586.0	672.0
at 30 knots speed, lbs.....	608.0	573.0	...
Five-Hours' Trial, at 24 Knots Speed.			
Mean speed realized, knots.....	25.56	24.74	24.0
Coal burned, total per hour, tons.....	6.45	5.12	4.8
per square foot G. S. per hour, pounds.....	61.5	48.0	46.2
mile covered during trial, pounds.....	560.0	460.0	444.0
at 24 knots speed, lbs.....	515.0	440.0	444.0
Eight-Hours' Trial, at 14 Knots Speed.			
Kind of machinery.....	3 Parsons turbines.	2 Rateau turbines. / 1 recipr. engine.	2 recipr. engines.
Total displacement, tons.....	470.0	468.0	424.0
Mean trial speed, in knots.....	13.872	14.09	14.083
Coal burned, per hour, pounds.....	2,142.0	1,421.0	1,010.0
square foot G. S. per hour, pounds.....	18.3	11.2	9.0
mile covered at 14 knots, pounds.....	144.0	100.0	72.0
Quantity of coal necessary to cover 1,170 miles, at a speed of 14 knots, tons.....	80.8	52.3	37.4



We find:

That between 10 and 25 knots the *Carabinier* with reciprocating engines does better than the *Chasseur* with Parsons turbines, and above 24 knots the *Voltigeur* with combination machinery does better than either of the other two. Below 24 knots the engine ship is slightly better than the combination boat and has a decided advantage over the ship with turbines only. Above 25 and 26 knots the engine ship is considerably to a disadvantage. Moreover, the combination ship made nine-tenths of one knot more on the same coal consumption than

the turbine boat and three and a half knots better than the engine boat. The displacement in the *Chasseur* and *Voltigeur* was about the same.

In backing and for maneuvering, with the boats, running at 20 knots, and after the signal "full speed astern" was given, the distance passed before a full stop was reached was for *Chasseur* 600 feet and for *Voltigeur* 500 feet. The engine ship stopped in about 450 feet.

The table appended gives a summary of the trials in a condensed form.

The following additional data concerning these and other French destroyers appeared in the "London Engineer," March 11, 1910.—H. C. D.

THE LATEST FRENCH TORPEDO-BOAT DESTROYERS.

The latest torpedo-boat destroyers delivered to the French Navy are of improved types. They may be divided into three classes: (1) Those fitted with reciprocating engines; (2) those driven by turbines; and (3) those driven by a combination of turbines and reciprocating engines. The boats in the first class have not come up to the expectation of their designers. The others look like some of the best foreign vessels of similar type, but they are smaller. Moreover, they are really nothing but experimental boats, and it is understood that the vessels of the future will be driven by that type of propulsive machinery which is found to give the best results in them. All the new destroyers vary slightly in design and size, as each builder had to submit his own proposals to the Marine for approval. In some cases these proposals had to be modified to meet the wishes of the French Admiralty, and it is asserted that these modifications did not improve the results achieved during trials.

The idea of the Marine was that the vessels should carry an armament of six quick-firing guns of 65-mm.—nearly $2\frac{1}{2}$ inches—diameter of bore, arranged three forward and three

aft, and be furnished with three torpedo tubes for 450-mm.—nearly 17¾ inches—torpedoes, one being arranged in the stem. Six torpedoes were to be carried. The speed was to be 28 knots for one hour and 25 knots for four consecutive hours. The radius of action at 14 knots was to be 1,170 miles. Four water-tube boilers placed in two separate compartments were to be provided, and the main engines were also to be contained in separate compartments. The diameter of the circle in which the boats were to turn was to be 400 m., and the vessels were to be capable of stopping in 250 m. when traveling at a speed of 20 knots. High-tensile steel, galvanized, was to be used for the hulls. Particular care had to be given to the design and construction of the hulls, which had to be strengthened in all places where special stresses are developed, and more especially under the guns and torpedo tubes. The outside plating had to be clincher riveted, and in the central part of the hull carvel work with inner butt strips had to be employed. The keels had to be 18 inches deep and about 141 feet long, with a normal thickness of 6 mm. The stem had to be of forged steel with lugs for attachment to the forward stringers. The stern and outboard propeller-shaft bearings had to be of cast steel, the rudder to be of the same material and covered with galvanized plates 2.5 mm. thick.

The vessels are divided into ten watertight compartments. These compartments are somewhat differently arranged in the various boats by different builders and by reason of the different types of propelling machinery. The first compartment—the fore peak—is very narrow and is empty. The second and third compartments are devoted to the crew, and contain the necessary lockers for clothes, &c. Below the second compartment is a fresh-water tank of a capacity of 3,000 liters. Below the third compartment is the chain locker and the store for the ammunition. The fourth and fifth compartments are, in the majority of cases, boiler rooms, the sixth compartment containing the propelling machinery, of whatever type it is. In those destroyers which are only driven by reciprocating engines the boiler rooms are situated both forward and aft of the

machinery space. The eighth compartment is allotted to the officers, and the ninth to the petty officers and chief engineer. Aft of this again is a storeroom in the after peak.

All the boats, especially those driven by reciprocating engines, are too low forward. They make bad weather and ship a great deal of water. Both the British and the German boats are considerably better designed in this respect. It is complained, too, that the officers' and crews' quarters are not what they ought to be, the consequence being that the personnel does not get a proper amount of rest, and is therefore not as efficient as it might be.

The boilers are either of the Du Temple or the Normand types, and the following are the leading particulars:—

	Du Temple.	Normand.
Grate area, square meters	22.60	21.40
Heating surface, square meters	1,028.0	920.0
Steam pressure per square centimeter, kilograms	18.0	18.0
Consumption of coal per square meter of grate (allowed), kilograms	400.0	400.0
Ditto, average during trials, kilograms	319 to 320	
Number of tubes per boiler	1,102	1,074
Internal diameter of tubes, millimeters	24 0	25.0
Outside diameter of tubes, millimeters	30.0	30.0

Vessels of the *Hussard* type have two sets of triple-expansion three-cylinder engines. The *Spahi* and *Carabinier* have two sets of four-cylinder triple-expansion engines. The *Voltigeur* and *Tirailleur* one set of triple-expansion three-cylinder engines and two sets of turbine engines. The *Chasseur* and the *Actée* are both driven by three sets of turbines.

The boats fitted with reciprocating engines only have given a good deal of trouble, and are by no means the fastest of the flotilla. Numerous preliminary trials were necessary in order to obtain the speed down to the last fraction of a knot stipulated in the contract. In this way there was much wear and tear of the engines before the vessels were taken over.

The *Chasseur* is driven by Parsons turbines—two cruising, two ahead and two astern. This type of equipment is so well

known that it is not necessary to speak at greater length about it.

The *Tirailleur* is the second boat of the French Navy to be driven by Breguet turbines. The first was the torpedo boat *No. 294,* which has now been turned into an instruction vessel. The main reciprocating engines of the *Tirailleur* run at 350 revolutions a minute, and is designed to develop some 3,000 indicated horsepower. Two independent sets of Tyrion oil pumps are used for the forced lubrication. The engine and its auxiliaries were built at the Chantiers de l'Atlantique of St. Nazaire. The contract for this vessel provided that up to 20 knots the steam, after having passed through the reciprocating engine, was to be led to the high-pressure and low-pressure turbines in turn, and thence to the condenser. At full power, with a boiler pressure of 18 kilos. per square centimeter, the turbines are estimated to give 6,000 indicated horsepower at 750 revolutions. There are, we may add, four turbines—two ahead and two astern.

The Breguet turbines are of the multiple-disc de Laval type. The blades are made of high-tensile bronze, and they are cold drawn, the length of the blades being in the same direction as that in which the metal was rolled. The fixed and moving blades are dovetailed into slots formed in the inner periphery of the casing and the outer periphery of the rotor, and are riveted in place.

There are two independent condensers, one for the reciprocating engine and the other for the turbines. They have 105 and 406 square meters of cooling surface respectively.

The *Voltigeur* and the *Actée* are perhaps the most interesting boats of the flotilla. They were designed by Monsieur Laubœuf, who is called the father of the French submersible fleet. The lines are peculiar, especially astern, where the vessels look like big motor boats, but the propellers are able to work in comparatively still water, and hence more efficiently. It is considered to be owing to this that the *Voltigeur* has made the best speed on trials of any of the destroyers built for the French Navy during the last two years. It is worthy of note

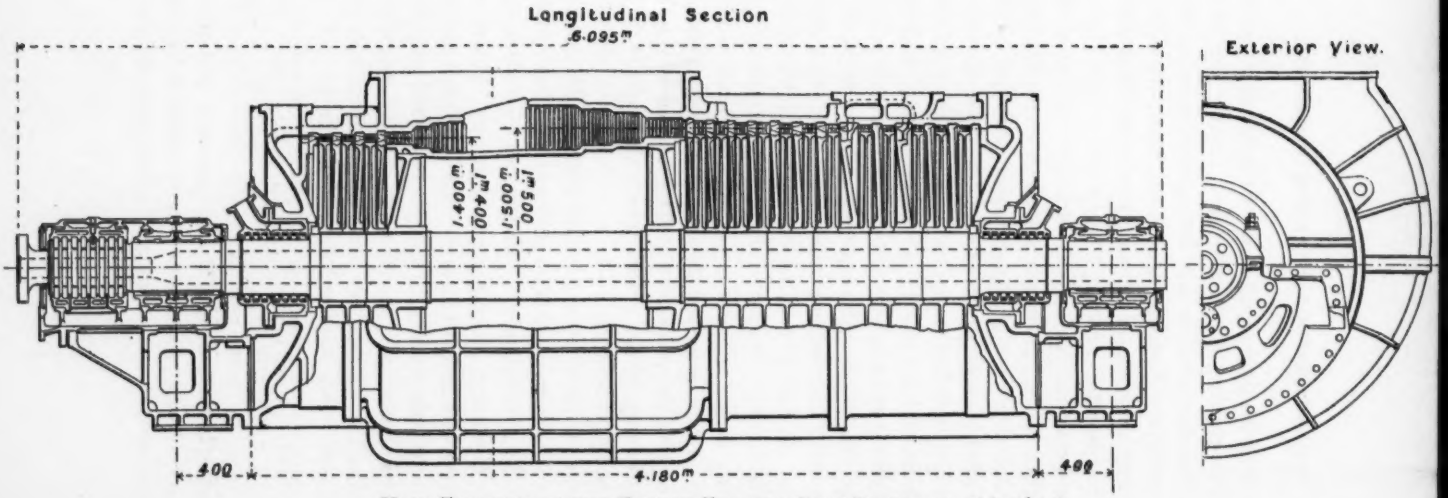
also that she is the first vessel in the French marine to be fitted with Rateau turbine machinery, and having regard to the speed she has developed, it will be interesting to see what the 750-ton destroyers *Fourche* and *Faux,* which are also to be propelled with this type of machinery without the assistance of reciprocating engines, will do.

The *Voltigeur* has six turbines arranged to drive two outboard shafts and a reciprocating engine on a middle shaft. On the starboard there is the high-pressure and one of the low-pressure turbines, both for ahead working. On the same shaft there is the low-pressure astern turbine. On the port side there are a medium and the other low-pressure ahead turbines, and also the high-pressure astern turbine. The steam is expanded in the high-pressure, medium-pressure and low-pressure turbines, exactly as in a four-cylinder triple-expansion reciprocating engine. At full power the turbines of the *Voltigeur* run at 900 revolutions per minute. At ordinary cruising speed the reciprocating engine alone is used, but the steam when it has done its work in it is passed through the turbines in sufficient quantity to drive the propellers at such a speed as not to interfere with the way of the vessel, which they might do if they were not made to revolve.

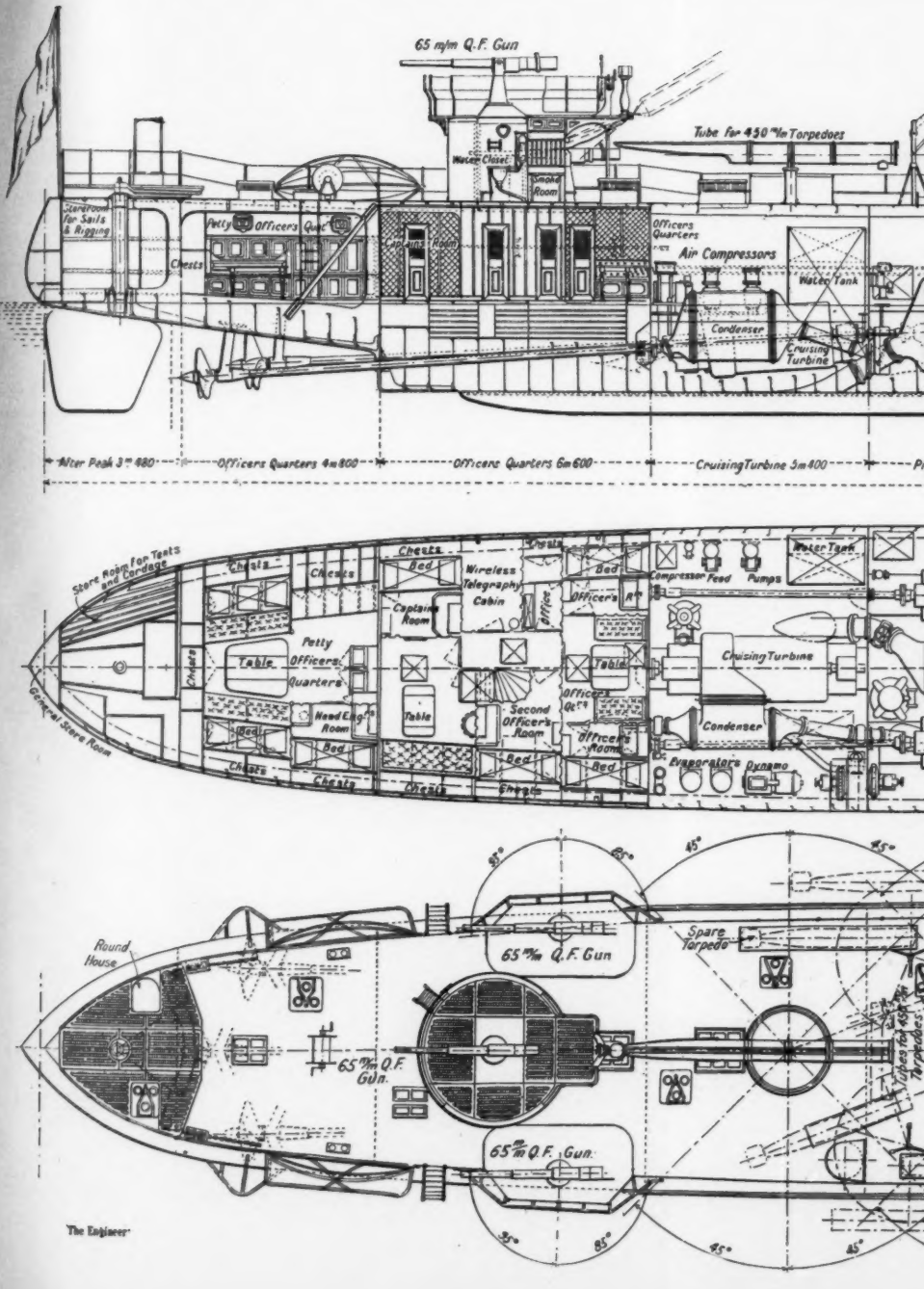
At the first trials there was some little trouble with the air pumps, but when these had been put right excellent results were obtained. The *Voltigeur's* condensers are on both sides of the turbines. The total cooling surface is 300 square meters. They are worked in conjunction with Westinghouse-Leblanc dry-air pumps, which are driven by 23-horsepower motors of the Boulte-Labordière type, running at 720 revolutions per minute. On the preliminary trials, with only one pump in operation, the following results were obtained:

Speed per hour (knots).....	19	20	24	26	28
Vacuum in condenser (mm.)....	750	730	745	740	736

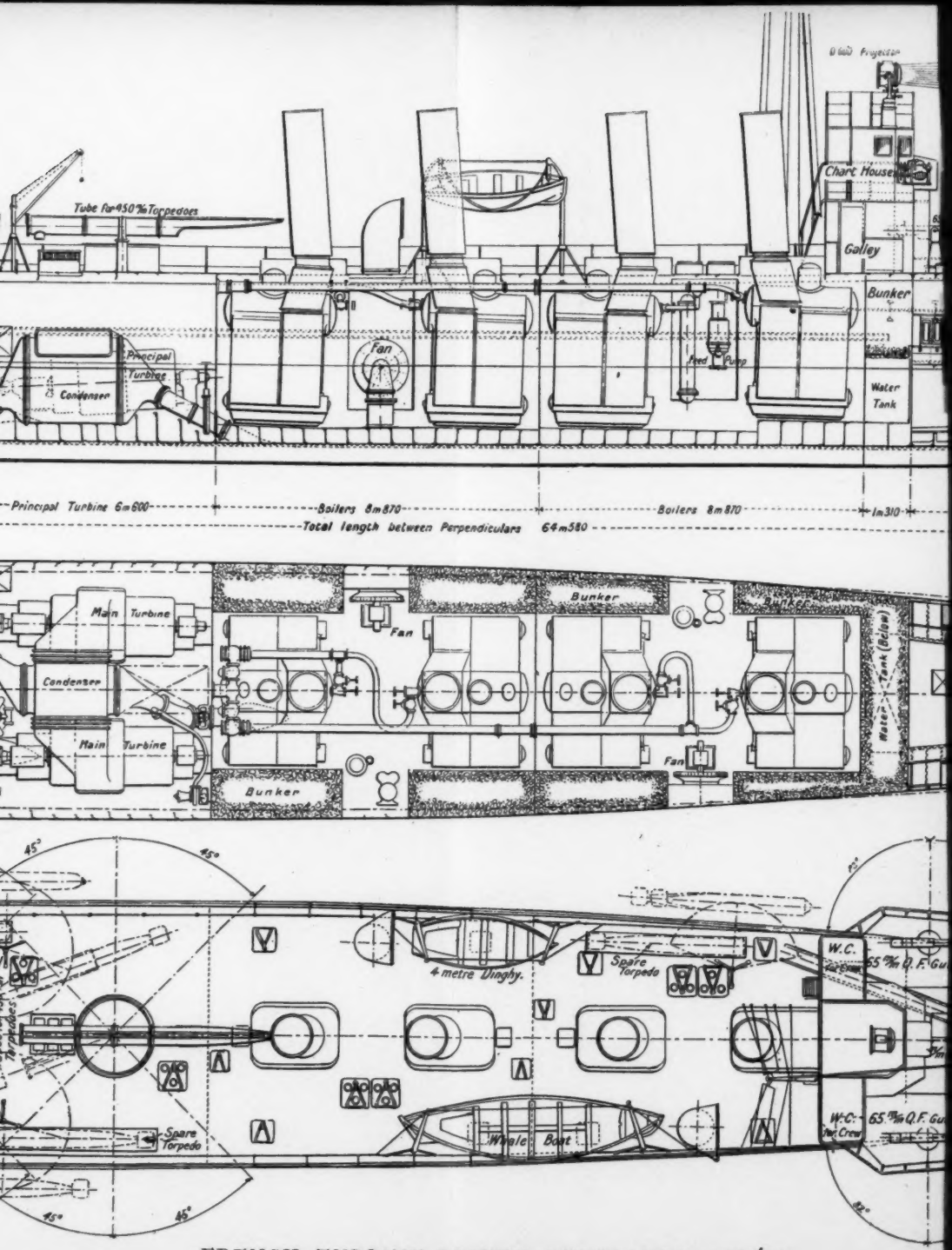
The atmospheric pressure at the time was 768 mm. The loss of fresh water was about 200 grammes per indicated horsepower, this being a usual amount. The total weight of the



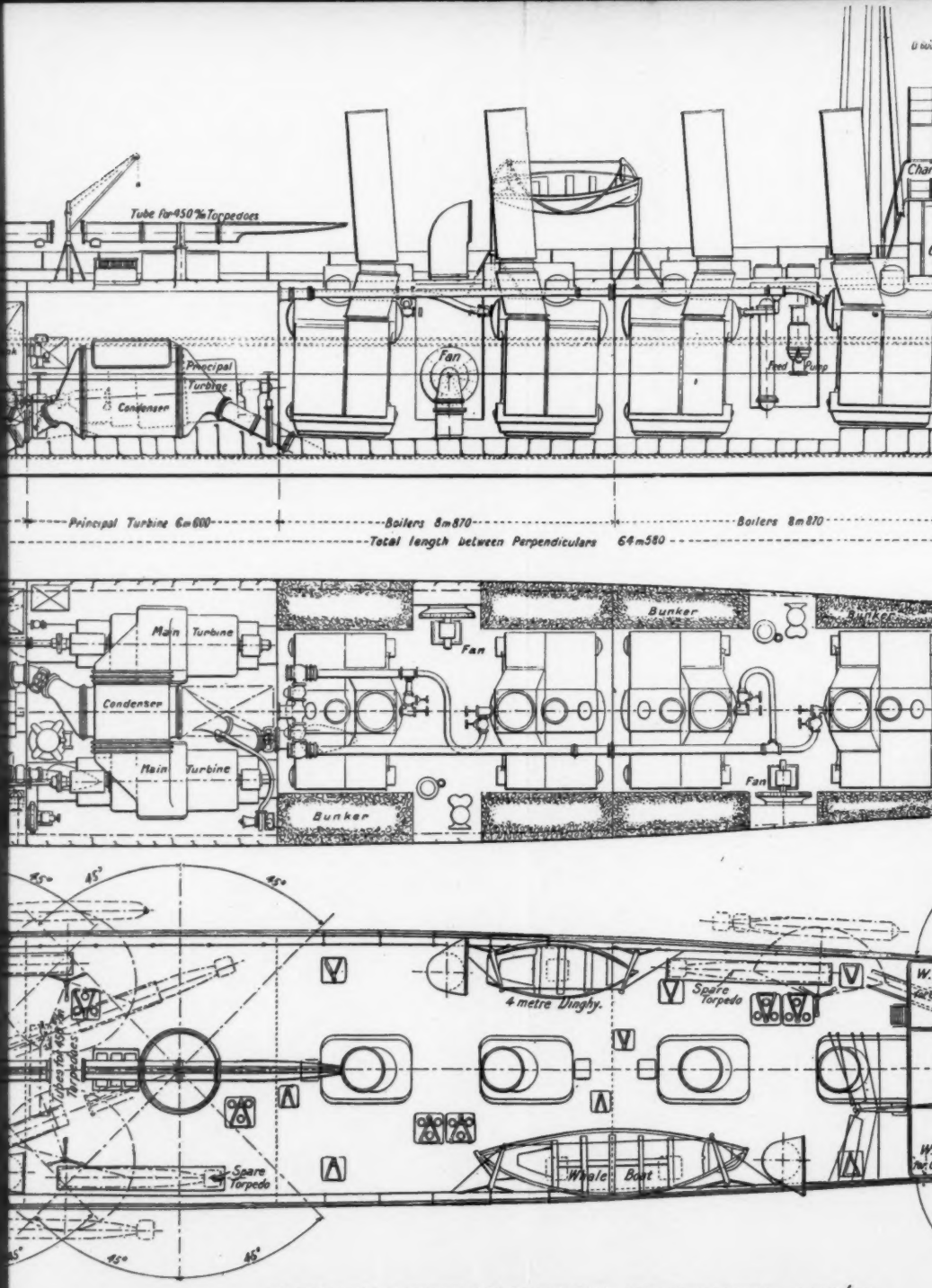
MAIN TURBINES OF THE FRENCH TORPEDO-BOAT DESTROYER "ACTÉE."



The Engineer



FRENCH TURBINE-DRIVEN DESTROYER ACTÉE.



FRENCH TURBINE-DRIVEN DESTROYER ACTÉE.

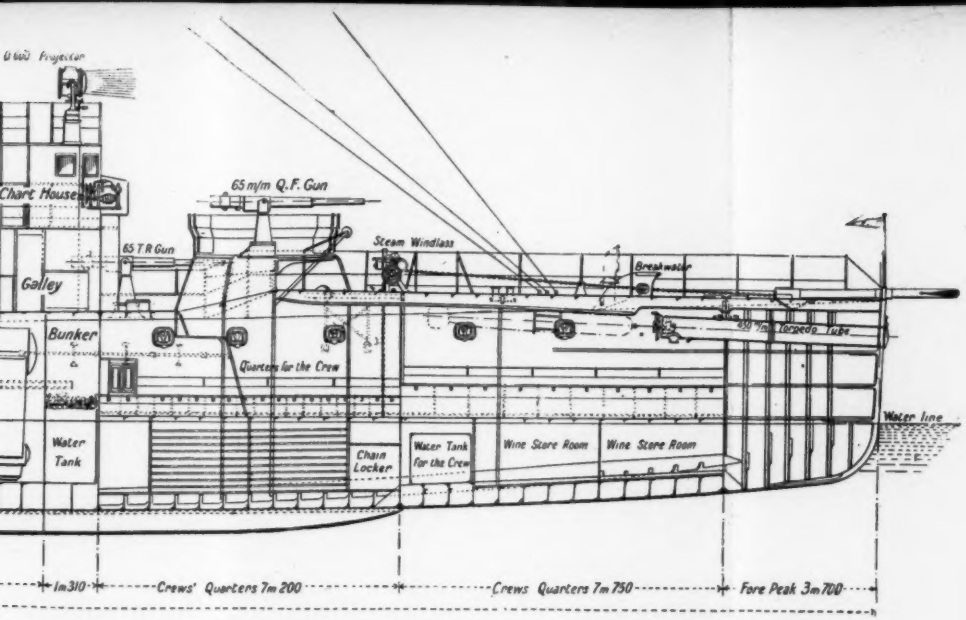
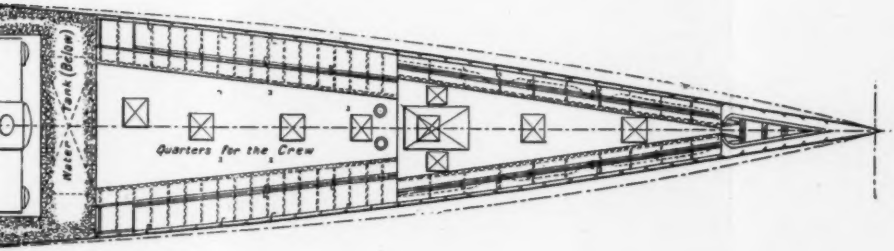
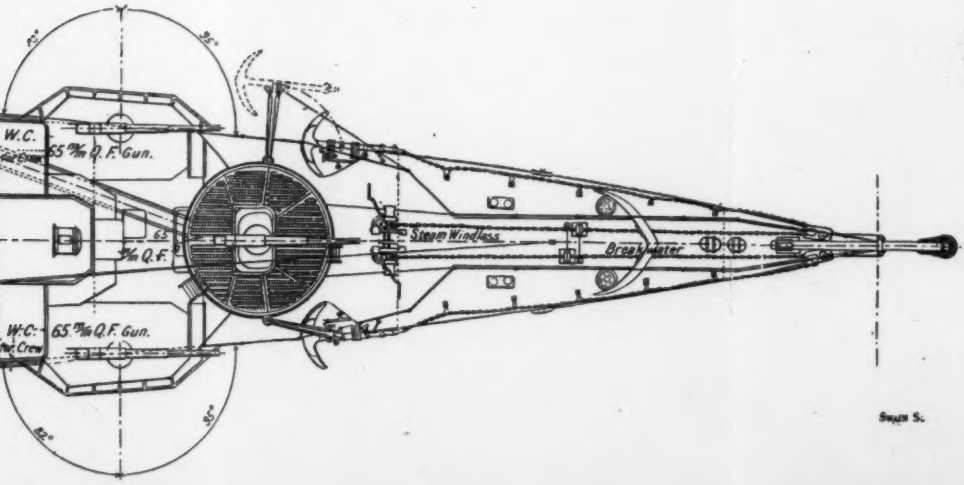


Chart House
65 m/m Q.F. Gun
65 T.R Gun
Steam Windlass
Breakwater
Galley
Bunker
Quarters for the Crew
Water Tank
Chain Locker
Water Tank for the Crew
Wine Store Room
Wine Store Room
Water line
1m310
Crews' Quarters 7m200
Crews Quarters 7m750
Fore Peak 3m700


Water Tank (Below)
Quarters for the Crew


W.C.
65 m/m Q.F. Gun.
Steam Windlass
Breakwater
W.C.
65 m/m Q.F. Gun.

turbines, without auxiliaries, was at the time of shipment 38,450 kilos., roughly 38¾ tons. They were built at the works of the Chantiers et Ateliers de Bretagne—late de la Brosse and Fouché—as were also the boilers, reciprocating engines, &c.

The 750-ton destroyers *Fourche* and *Faux* will, as mentioned above, be driven by Rateau turbines only. They will have three shafts, and it is expected that the numerous improvements which have been introduced will enable them to run very economically when at cruising speed. Their guaranteed full speed is 32 knots, and if they do as well as the *Voltiguer*—and it is fully anticipated that they will do so—this will be exceeded by 3 knots.

The destroyer *Actée,* which was built by Schneider & Co. in their yard at Creusot, is provided with three shafts driven by Schneider-Zoelly turbines of the combined type. We are enabled to give drawings of this vessel in our two-page Supplement, and of her main turbines. Each ahead turbine contains a high-pressure and a low-pressure stage, and each is, therefore, self-contained, and it and its shaft may be driven independently of the others. The astern turbines are the same as the ahead, only not so powerful.

One of the distinguishing features of these turbines is the small number of pipes and connections, as compared with those of other engines of a similar type. The turbines are to develop a total of 8,000 indicated horsepower. We may add that for the destroyer *Boutefeu* a set of engines of like design is being built. They are to develop 1,400 indicated horsepower, while those for the *Commandant Bory,* which are also of this type, are to indicate as much as 16,000 horsepower. In both these vessels there are only to be two shafts.

The steering gear of the *Actée* is in the chart house. It is of new design, requires but a small amount of power and works very well. Near the aft 65-mm. gun there is also a hand-steering wheel. The towing arrangements are similar to those ordinarily provided for vessels of this type. She carries one whale boat 7 m. in length, and one "you-you," or dinghy, made

of wood. These are hung on davits. There is also room for two collapsible canvas boats, 3.6 m. in length.

Electricity is supplied by a steam-driven dynamo capable of giving 100 ampères at 80 volts. In some of the other vessels the dynamos are driven by paraffin engines. The *Actée* carries six 450-mm. torpedoes.

TABLE No. I.

.....	*Hussard.*	*Carabinier.*	*Spahi.*
Length over all, meters	69.365	64.5	64.6
between perpendiculars, m...	65.8	64.2	64.6
Extreme breadth, meters	6.0	6.2	6.65
Depth, meters	4.1	4.18	4.5
Draught astern, meters	3.0	2.29	3.0
Displacement, loaded, tons	412.848	415.47	413.0
Boilers	4 Du Temple	4 Du Temple	4 Normand
Grate area, square meters	21.4	21.4	22.6
Heating surface, square meters	920.0	920.0	1,028.0
Working pressure per square centimeter, kilos	18.0	18.0	18.0
Air pressure in stokehold	From 65	to 75 mm.	of water.
Bunker capacity, tons	120.0	110.0	106.0
Number of cylinders	3	4	4
Diameter of cylinders, millimeters	550, 720 and 1,230	510, 700, 870, and 870	540, 840, 920, and 920
Stroke, millimeters	500.0	500.0	650.0
Revolutions per minute	305 to 310	333.0	295 to 313
Horsepower, contracted for	7,000.0	7,500.0	8,200.0
on official trials	7,760.0	8,200.0	9,600.0
Propellers, number	2	2	2
Diameter and pitch of propellers, m.	2.45 and 3.4	?	2.6 and 3.56
Radius of action, knots	1,170 at 14 k	nots ; 280 at	full power.
Armament	6 65-mm. gu	ns and 2,100	shells ;
	3 torpedo tu	bes and 6 tor	pedoes.
Crew	65	?	60
Speed, contracted for, knots	28.0	28.0	28.0
on official trials, knots	29.67	27.41	28.05

The lessons learnt from these boats are, first of all, that the freeboard is too low. It should be at least from 4 m. to 5 m. at the stem. In later vessels the hull is to be reinforced under the gun mountings. Moreover, it is considered that the 65-mm. guns are too small, especially for destroyers, which are to range up to 900 tons. It has been appreciated that the limit of speed possible with reciprocating engines has been reached, and as foreign destroyers, which may have to be encountered in time of war, have speeds of 33 knots and upwards, it is thought

absolutely necessary that all future destroyers should be fitted with turbines, as they alone can give the high speeds which will be required to bring the French on a parity with other nations. It is being strongly urged that oil fuel should be used in all future vessels of this type, on the grounds that it is much

TABLE No. II.

.....	*Chasseur.*	*Tirailleur.*	*Voltigeur.*	*Actée.*
Length over all, meters	67.65	...	...	...
between perpendiculars, meters	64.2	63.0	65.6	64.58
Extreme breadth, meters	6.75	6.84	6.96	6.76
Depth, meters	4.5	?	4.37	4.52
Draught astern, meters	2.82	2.9	2.65	...
Displacement, loaded, tons	497.5	414.0	445.0	470.0
Boilers, number and type	4 Normand	4 Du Temple	4 Normand	4 Normand
Grate area, square meters	21.4	22.6	21.4	21.4
Heating surface, square meters	1,020.0	920.0	1,028.0	1,028.0
Steam pressure per square centimeter, kilos	18.0	18.0	18.0	18.0
Air pressure in stokehold	65	to 75 mm.	of water.	
Engine, type	...	triple-expansion	triple-expansion	...
Diameter of cylinders, mm	...	500, 720, and 1,000	...	...
Stroke, millimeters	...	500.0	...	...
Revolutions per minute	...	350	...	...
I.H.P., contracted for	...	...	...	...
on official trials	...	3,000.0	...	...
Turbine's, type	Parson's	Breguet de Laval	Rateau.	Schneider-Zoelly
Number, ahead	4	2	4	2
astern	2	2	2	2
Diameter, cruising, meters	1.2	...	...	...
H.P. ahead, meters	1.2	1.1	...	...
L.P. ahead, meters	1.6	1.707	...	...
H.P. astern, meters	1.6	1.4	...	...
L.P. astern, meters	1.6	1.707	...	...
Revolutions per minute	940	750	...	...
Condenser surface, square meters	...	6.0	...	...
Vacuum at full speed, mm	720.0	...	...	...
Number of propellers	3	3	3	3
I.H.P., contracted for	7,200.0	7,800.0	...	...
on trials	8,600.0	?	9,050.0	...
Radius of action	1,170.0	1,170.0	1,170.0	...
at full speed	300 0	...	...	...
Armaments	6 guns of 3 torpedo	65 mm. bore, 2,300 tubes, 6 torpedoes.	450 mm. dia.,	shells;
Crew	58	...	65	63

easier to obtain with it a continuous high speed, to get the best work out of the boilers, and to do it without smoke. It is further put forward as a reason for this action that the number of stokers may be reduced.

We give herewith two tables which contain the leading particulars of the *Hussard*, the *Carabinier*, the *Spahi*, the *Chasseur*, the *Tirailleur*, the *Voltigeur* and the *Actée*.

THE LATEST MARINE ENGINE PROPOSITION.

THE HERRICK ROTO-TURBO COMBINATION.

By Captain A. B. Willits, U. S. N., Member.

In presenting these notes on a new and remarkable combination which, by its peculiar fitness to the compelling wants of present marine-engine conditions, appears to provide the means for reaching the Ultima Thule in economical steam installations, a brief resumé of antecedent developments is an advisable preface.

Despite the years that have passed since practical steam engineering began its existence as a profession, and notwithstanding the numberless books that have been written and read, bearing upon the properties of steam, the functions of the steam engine and the mechanical equivalent of heat, the vast majority of engineers devoting thought to the improvement of the apparatus for transforming thermal units borne by the steam, into useful work in the motor, have only studied the structural features thereof and have brought their ingenuity to bear mainly upon perfecting the form and dimensions of the different parts and the proper distribution of the loads.

Looking back to the experiments of Joule, some sixty years ago, the progress has been slow, and the decided steps comparatively few, in the direction of capturing an increased number of B.T.U. for mechanical effort per pound of steam used and to reducing the ever great loss to a nearer approach to that inherent in the heat engine and fixed by Thermo-dynamic law.

Periodically some deeper thinker has demonstrated an ad-

vance along the lines of economy, but during the past thirty years the principal endeavor in Marine Engineering has been to devise machinery installations which would secure the increased horsepower demanded for desired speeds without that enormously prohibitive increase in size and weight which would follow from designing a "sufficient" simple engine.

PROGRESSIVE STEPS IN DESIGN.

This course naturally began by increasing the initial pressures used, and in order to expand this higher-pressure steam to a reasonable terminal pressure, and at the same time to avoid abnormal distribution of load and effect on the crank shaft, the division of the work in sequence cylinders was obviously necessary; and the actual economy resulting from the use of high pressures and multi-expansion engines was less an inspiring motive than a pleasing consequence.

Full consideration of the steam engine in its development as a heat transformer has never been general, and it is only since the advent of the turbine that a closer concern has been manifested throughout the engineering world, in tracing the mechanical equivalent of the thermal units through the engine and reconciling, as far as possible, the theoretical output, by heat drop, with the actual as obtained by card.

Certain gain in economy by use of superheated steam and increased cylinder ratios are only now being proven in installations on two of our own recent battleships, just when the reciprocating engine seems to be disappearing, and the results already obtained show astonishing efficiencies.

Recently, however, and consequent upon the recognition of the merits of the turbine, a growing restlessness has invaded the engineering field. A general awakening to the possibilities of increased thermal efficiency has given an attractiveness to the profession never before excelled, and has concentrated the thoughts of the ablest designers upon the curtailing of heat losses, in both the main and auxiliary elements of marine machinery, to an extent as promising as it is interesting.

The results of trials of the earlier Parsons marine turbines indicated that, at full powers, this installation was very considerably more economical than reciprocating engines at designed speed; and while this phase of supremacy is now challenged by the performances of the *Michigan* and *South Carolina,* there is no doubt but that this promise of economy then extended gave an initial impulse to the venture of the Cunard Line in installing 70,000 B.H.P. Parsons turbines in each of the two greatest express steamers in the world, the *Mauretania* and *Lusitania,* three years ago. Apart from the design lending itself admirably to the installation of such enormous power within a reasonable limit of space and weight, the conditions of the mail-carrying service made it entirely unnecessary to consider any but full speed as a desirable factor.

THE APPEAL OF THE TURBINE.

But there were also other features besides that of economy, possessed by the new type, which appealed forcibly to the engineer, and without which it is not probable that any company would have dared such a financial risk as accompanied this radical departure from the known and tried type of marine machinery, and its tremendously long stride forward from the highest powers at that time in service in marine turbines. Engineers throughout the world were aroused to keenest interest and doubt stood hand in hand with hope. Here was an experiment on such a grand-opera scale as to make its success or failure a matter of most signal value to nations as well as corporations.

In the opinion of the writer, the results of these trials announced the doom of the reciprocating engine! Not that the knock-out blow was then and there delivered, nor that reciprocating machinery would cease to be installed in steam ships, but, from the standpoint of sound engineering, the outlook for the future then became one wherein the converging lines of perspective met at the ideal type for maintaining continuous motion—*an all circular path for the moving parts*—approach-

ing the scheme of nature in the great machinery of the planetary system; and the only method by which the transmission of power can be kept free from the shock and ill effects of constantly-repeated reversal of the direction of travel of the heavy transmitting elements.

This convincing establishment of the economical possibility of the turbine for marine work was not inclusive of installations which were intended to operate at greatly varying speeds, for it was early demonstrated that the economy of the turbine only exceeded that of the old type of marine engine at or near full power, and fell seriously below this at the reduced speeds necessary to reserve to war ships to obtain the largest possible "cruising radius"—the greatest possible distance of travel per ton of coal carried.

It must be conceded, therefore, that the strongest appeal made by the turbine to naval engineers was through these other features referred to above rather than that by superior economy, for the new type at once began to supplant reciprocating engines in the new vessels of great Naval Powers. These other features lay in the remarkable structural changes brought by the turbine, and by which a multitude of practical troubles were at once eliminated. It is scarcely permissible to enumerate these changes, so rapidly have they become known to all familiar with the marine reciprocating engine, with its massive overhead cylinders and valve chests, its numerous joints, "journals" and connections, its complex lubricating system and that eternal wear of brasses with the noises and losses due thereto. All of these were sources of continual anxiety and demanded constant watchfulness and work to keep the whole in readiness for the emergency call for sustained full speed.

With the coming of the turbine these troublesome characteristics all disappeared, and in the substitute type the operation was a miracle of smoothness and noiselessness, and whatever few points it possessed liable to derangement were at least not in evidence.

DEFICIENCIES OF THE TURBINE.

But with all the many advantages attendant upon the use of the turbine, it is impossible to ignore its failings, and the question of how to retain it without paying too much for its output at reduced speeds, agitates the professional mind at present.

We are shown that the great point of superior economy of the turbine over other steam engines lies in the low-pressure end, and practically secures there a comparatively enormous amount of useful work from that part of the steam which, in the reciprocating engine, expands from the terminal of low-pressure expansion to the tension of the condenser without work.

Practically, with present high boiler pressures, we can only expand our steam in the L.P. cylinders of reciprocating engines to about 12 or 13 pounds absolute, at terminal of expansion, and owing to the necessarily restricted passages of exhaust ports we have always a back pressure to work against considerably higher than that corresponding to the tension of the vapor in the condenser itself. This back pressure averaging about 6.7 pounds absolute, 16½ inches vacuum, as noted by Engineer Vice-Admiral Henry J. Oram, C. B., of the British Navy, in his admirable and comprehensive address to the Junior Institution of Engineers last November. In the turbine, however, we can secure useful expansive work on the blades down to within about ½ pound of the absolute pressure in the condenser, and by maintaining a lower tension in the latter than is customary in reciprocating plants, we garner additional power far out of proportion to the increased vacuum, *per se*. This is obtained by the full area afforded in the turbine for the normal increase in volume of the steam as it passes to the lower pressures, permitting its velocity to flow unchecked by such restrictions as are existent in the piston engine, and securing the useful work on the blades due to the heat drop in these last stages. This drop bears an astonishingly large relation to the total heat output, and Vice-Admiral Oram points out in his statement that the B.T.U.'s given up by the steam

expanding adiabatically, from the 6.7 pounds absolute pressure (referred to above as the average least back pressure in L.P. cylinders of reciprocating engines), down to the tension corresponding to 28-inch vacuum, amount to 38.7 per cent. of the B.T.U.'s given up in the adiabatic expansion of the steam from 235 pounds absolute *down to* this 6.7 pounds absolute.

From this we can see that if it was possible to obtain an equal efficiency in the upper stages of the Parsons turbine with that of the reciprocating engine we would gain far more than we now do, per pound of steam flow, by using all-turbine installations. But such economy in the upper stages of the turbine are wholly impracticable. Here, for high pressures, the blading is short, the leverage or drum diameter is least, the clearances are (proportional to blade area) greatest and the peripheral velocity limited, of course, by the conditions of marine work and propeller efficiency.

THE ALL-IMPORTANT PROBLEM.

From the test of several Parsons turbine outfits used on the British battleships we find increasing efficiency, or a decreasing water rate per B.H.P. in those installations in which decreased initial pressure was designed for and used; thus giving a greater possibility of efficiency in the upper stages by more effective blading and larger drum diameters. But to recede from the position we have gained after such tedious efforts, in securing and controlling high-pressure steam, would scarcely be countenanced, and we are forced either to face the problem of a combination, in which we can utilize an efficient high-pressure unit for our upper expansions and exhaust into turbines for the lower expansions, or else give rein to unlimited speed of rotation of turbines alone (and thus increase our total economies), and transfer its power, electrically or by gearing, to a low-speed propeller shaft.

This latter scheme of "gearing down" is actually in course of application to one of the new colliers building at Baltimore, Maryland, and the actual results in points of efficiency, econ-

omy, wear and general adaptability are awaited with much interest. The use of gearing on the main drive has never been a popular idea for marine work, but the proof of the pudding being now within sight, it would be a foolish critic who would proclaim his conjectures.

As to the first alternative of interposing some unit, other than turbine, between the boiler pressure and the turbine in which to utilize the upper expansions economically and deliver steam of a most suitable pressure for economy to the turbine, we are offered combinations with the reciprocating engines which show absolute certainty of much lower water rate, at variably reduced speeds, than that obtainable from either type alone. The *Otaki*, of the New Zealand Ship Co., was the pioneer in this installation, using a three-screw arrangement and driving the wing screws by triple-expansion reciprocating engines, the exhaust from which being used in a L.P. turbine driving the center shaft.

The results were excellent but the "type characteristic" of the propelling machinery is a compromise which practically obscures those advantages of circular motion, which we now refuse to ignore. Here we retain reciprocating machinery with all its old inherent faults and with the difficulties of handling highly superheated steam still to be maneuvered. For it must be remembered that we are to use superheat in the high-pressure units of a desirable combination in order to not only realize its economy therein but also to deliver dry or saturated steam to the turbine; and while we are using a fair degree of superheat in the two battleships noted above we are not yet beyond the service experimental stage in so far as knowledge of best materials and efficient control are concerned.

The recipro-turbo combination is therefore far from ideal, and we must look for a better form and make sure that it is unobtainable before accepting this compromise as the new standard.

The electrical-transmission scheme also lacks many desirable attributes, and while attractive in the abtsract brings with it a world of uncertainties as to its sufficient flexibility and gen-

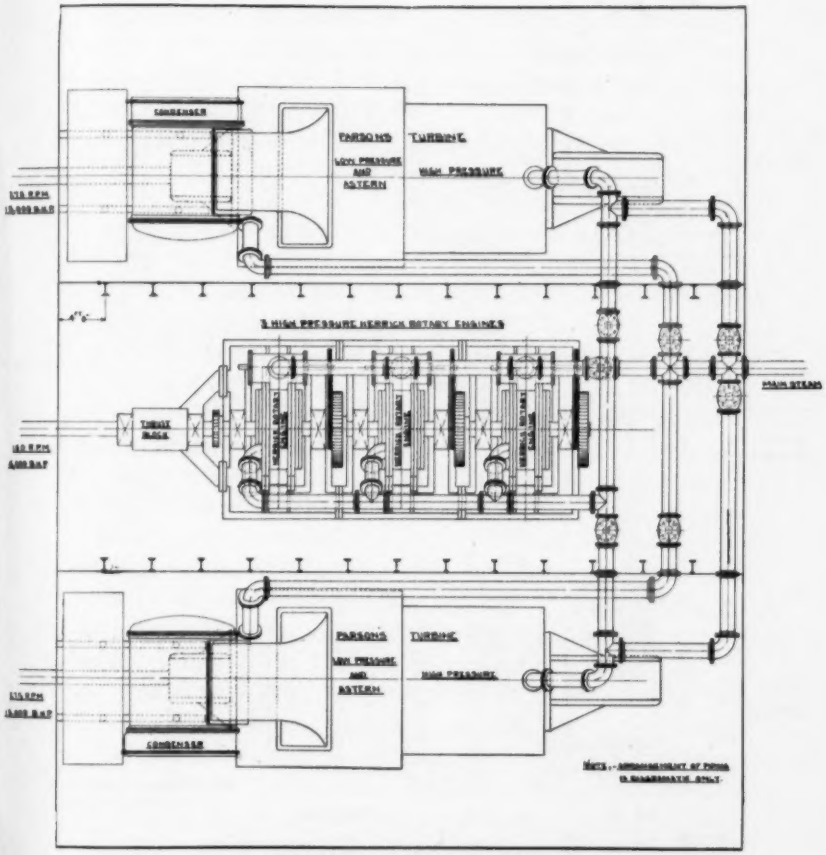
eral adaptability to the main drive of a big cruising ship. In its present stage it certainly lacks the maturity necessary to inspire confidence, nor, even if capable of development into a "grown up" of the physical attainments predicted for it by its parents, would it be capable of holding its own in all points against the latest arrival to which we are leading the reader through the foregoing *resumé*. Strange to say, in view of the long-discredited type which it follows, the new comer claiming recognition as the rightful heir to the seat of power of the upper stages of an ideal combination is a Rotary Engine, devised and improved through years of laborious devotion, by Mr. Gerard P. Herrick, of New York.

THE ROTARY IN AN IDEAL COMBINATION.

Usually when an inventor broaches the subject of Rotary Engines, in engineering circles, a sympathetic smile hides the *dis*interest excited, for the prejudice of years, due to the failures upon failures of attempts to perfect this theoretically-ideal motor, has led to the classifying of the Rotary as a Forlorn Hope.

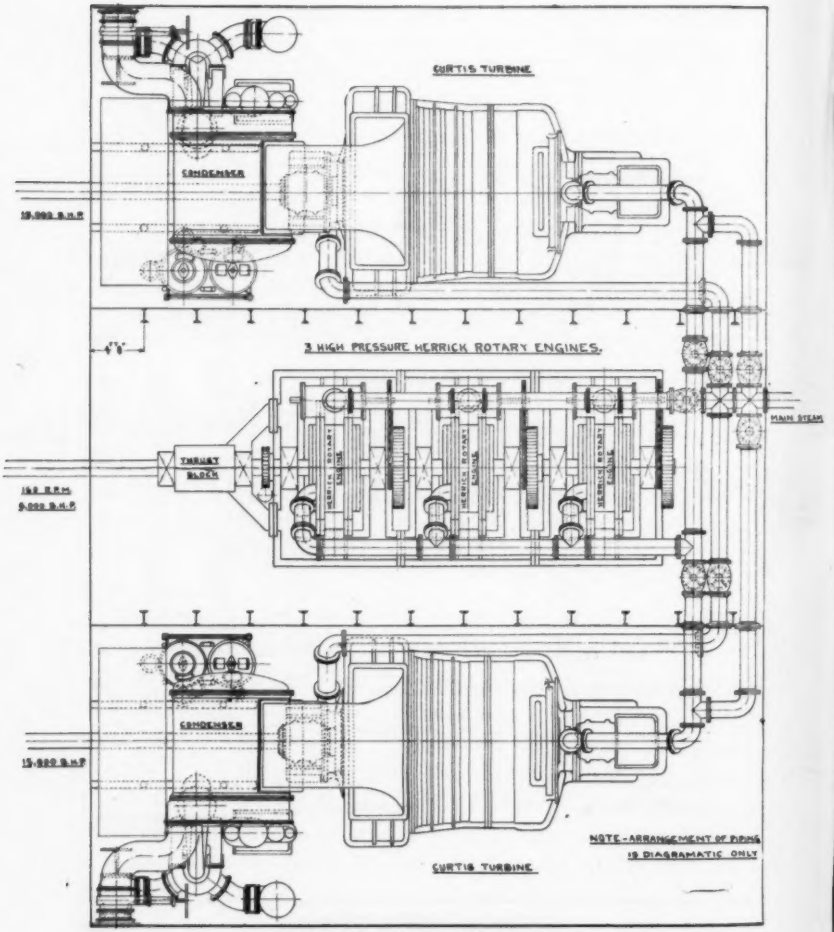
The great difficulties in the way of producing a practical and economical motor of this type were those of balancing the radial loads and packing the rotating parts effectively and without excessive friction; but when it is seen that, if, in a Roto-Turbo combination, such as we have indicated, steam does leak past the piston (provided it be not abnormally out of proportion to the total flow), it will simply be utilized at a low-water rate in the turbine, we are free to minimize friction at piston packing and to discover in a rotary engine with balanced radial loads the desired and requisite reducing unit for securing our intermediary power between boiler and turbine.

The automatic belancing of radial loads together with an entire avoidance of actual contact of the piston with the walls of the casing, while yet retaining such close control of these clearances as to reduce leakage to the point of minor significance, are the essential accomplishments of the Herrick Rotary. Its



PROPOSED ARRANGEMENT
OF
36,000 B.H.P MARINE PROPELLING INSTALLATION
HERRICK ROTARY ENGINES IN COMBINATION WITH 2 OF PARSONS STEAM TURBINES
DATE - 18 - APRIL - 1910

Fig. 1.



PROPOSED ARRANGEMENT
OF
36,000 B.H.P. MARINE PROPELLING INSTALLATION
3 HERRICK ROTARY ENGINES IN COMBINATION WITH 2 CURTIS STEAM TURBINE
DATE - 18 APRIL - 1910

Fig. 2.

appearance is as remarkable in its timeliness as the device itself is in its adaptiveness to our pressing needs in turbine combinations, and as the roto-turbo combination is one wherein steam of the highest obtainable pressure and superheat can be safely handled, and the desired moderate pressure, most suitable for the turbine, delivered to the latter in saturated form, we have assurances of such serious reduction of water rate per unit of total power output as to bring the scheme at once into the field of demand for future installations.

Before describing in any detail Mr. Herrick's invention let us look for a moment at this obvious economy, temporarily putting aside, if we can, the gratification we experience in realizing that we are now able to retain all circular motion with its manifest mechanical advantages.

Mr. Luther D. Lovekin, Chief Engineer of the New York Shipbuilding Company, while in consultation with Mr. Herrick regarding the field for the balanced rotary, first pointed out its singular fitness for this upper-stage unit, and concluded from calculations based upon actual marine turbine performance that a water rate of 10 pounds per B.H.P. would be entirely within the possibilities of such a combined system.

In this he assumed a three-shaft installation having either Curtis or Parsons turbines of 15,000 B.H.P. each, on wing shafts, and positively capable of producing that power on a water rate of 13 pounds per B.H.P., with initial steam pressure of 140 pounds absolute; the consequent steam flow being of course 30,000 × 13, or 390,000 pounds per hour.

Upon the center shaft he proposed to place a Herrick Rotary and to use steam pressure in the Rotary of about double that designed for the turbines; and, while maintaining the same steam flow as before per hour, expand this high-pressure steam usefully in the rotary first, before delivering it at a normal pressure of 140 pounds to the turbines, for the development of the 30,000 B.H.P. therein.

The calculation for size and power of this center-shaft unit are as follows:

Assumed r.p.m. of rotary, ...	160
effective stroke of rotary, feet, . .	17
clearance (from general design) per cent., ...	12
Stroke, plus clearance, feet, ...	19
Distance swept by piston of the rotary per hour, allowing for clearance, will be $19 \times 160 \times 60$, feet, ...	182,400
Total cubic feet of steam used per hour (steam flow being 390,000 lbs. per hour and volume of 1 pound of steam at 140 pounds absolute being 3.2 cubic feet) is $390{,}000 \times 3.2 =$.	1,248,000
Area of piston, therefore, will be:	
$\frac{1{,}248{,}000}{182{,}400}$ = square feet, ...	6.85
Assumed, height of piston = inches, . . .	16
Width of piston $= \frac{6.85 \times 144}{16}$ = inches, . .	61.65

But for considerations of size and proper balance as well as in order to give that desirable steady flow of exhaust from rotary to the turbines, three separate rotaries (not compounded) are used, having their pistons 120 degrees apart, so that instead of one piston 16 inches high $\times$ 61½ inches wide we have three pistons, each 16 inches $\times$ 20½ inches, which will expand the 390,000 pounds of steam per hour from 280 pounds absolute to 140 pounds absolute.

In this expansion the work done by the rotaries will be as follows:

M.P., expanding 2 times, = 0.8465 of initial pressure.	
Theoretical M.P. 280×0.8465 = pounds, . . .	237
Back pressure, pounds,	140
M.E.P., pounds, . .	97
Ratio of actual to theoretical M.E.P., ...	.80
Actual M.E.P., pounds,	77.60

Assuming this to be 78 pounds, as not at all excessive, the indicated horsepower, therefore, using the total piston area, will be:

$$\frac{78 \times 17 \times 6.85 \times 144 \times 160}{33{,}000} = 6{,}380,$$

which, in a frictionless-piston engine, will be equal to about 6,000 B.H.P. secured on the center shaft at slow r.p.m., and with the consequent higher propeller efficiency.

The total output now becomes:

30,000 B.H.P. turbines, 50 per cent. propeller efficiency.
6,000 B.H.P. rotary, 60 per cent. propeller efficiency.

36,000 B.H.P.

While the water rate is now:

$$\frac{390{,}000}{36{,}000} = 10.8 \text{ pounds per B.H.P. per hour.}$$

Considering the entire fitness of the rotary for use of superheat (as will be shown later on) we can readily count on a still higher efficiency by superheating our steam far beyond that which is merely necessary to insure a dry exhaust from the rotaries or saturated steam for the turbines, and in following this course we are obviously within limits of reasonable expectation in assuming a resultant 10 pounds water rate for B.H.P. per hour, as finally attainable.

DETAILED ECONOMIES.

To the lay mind, the statement that the water rate can be reduced from 13 to 10 pounds per B.H.P. per hour bears with it no graphic picture of the tremendous economies involved therein. But when he is told that it means a 23 per cent. reduction in amount of coal used on any voyage he is very likely to sit up and take notice.

Obviously such a reduction in fuel required for propulsion of a ship would immediately operate to reduce the size and weight of the boilers for any given power, and this pregnant

feature must not be lost sight of in considering the advantages of the change from present installations.

Consequent upon this reduction in the size of the boiler plant there naturally follows an increased space available for stowage of fuel, so that, taking any battleship now in service, and contemplating the replacement of its machinery by the roto-turbo construction, we would have not only a great increase in cruising radius due to the smaller amount of coal per hour necessary to burn to secure a given speed, owing to superior engine economy, but we would also have the additional increase in cruising radius due to the larger amount of fuel normally carried.

Again, the question of cost is universally appealing, and not only will there be for any given power a greatly reduced first cost, or annual interest on investment, but a cash saving in daily fuel account when steaming, amounting, in cases of big ocean liners using 800 or 900 tons of coal per day, to probably four hundred dollars per day per ship.

Acknowledging that we are in the position of actually requiring an additional upper-stage unit to reduce the power cost of marine propulsion, and that the reciprocating engine (in spite of its long and faithful service and the embodiment it represents of such a vast amount of thought, ingenuity and experience of the ablest engineers) is, by reason of its necessary deficiencies, an "undesirable citizen" in future outlook, we are brought to the conviction that in the rotary exists the only possible satisfactory link between the turbine and the high pressures we can now obtain and handle safely in marine boilers.

And unquestionably *we must use high steam pressures and superheated steam* if we are to advance in economy and speed of ships without enormously increasing the weight and size of our machinery. The turbine has come to stay for that utilization of the lower expansions absolutely impossible in a reciprocating engine, and in this rotary we find the natural protector of the turbine without going out of the family.

A summary of all these advantages in a roto-turbo combination may be stated thus:

1. Highest obtainable steam pressures and superheat utilized and lowest pressures at terminal of expansion obtained.
2. All circular path of moving parts.
3. Loss due to friction of piston packing avoided.
4. Loss from back pressure due to contracted exhaust passages and gradual opening of exhaust valves absent.
5. No receiver necessary to secure steady flow of exhaust from upper unit to turbine.
6. Simplest in construction.
7. Cheapest to build and maintain.
8. Least number of attendants necessary to operate.
9. Absence of vibration.
10. Ideal in balance and steadiness of propulsive effort.
11. Low R.P.M. of upper unit and its propeller shaft.

Of course this is a new proposition involving work in design and actual use before many, besides those from the "Doubting State," will fall into line with the acclaimers of its merits; but to secure instantaneous appreciation of these advantages by professional engineers, no mental surgery is requisite.

THE HERRICK BALANCED ROTARY.

Bringing the rotary engine into practical and efficient form at this time also brings it at once into a field which engineering progress had almost miraculously prepared for it, and for which it had never even been schemed. Its designed usefulness lay in the smaller powers, while its incidental or accidental application to the combination magnifies its importance beyond the dreams of its designers.

Let us look now at the Herrick Rotary and see whether we can find in it a guarantee of stable efficiency. Figs. 3 and 4 show a section and elevation of this device as designed for the 36,000 B. H. P. marine installation used in Figs. 1 and 2 and in the calculations in the foregoing pages, and while illustrating

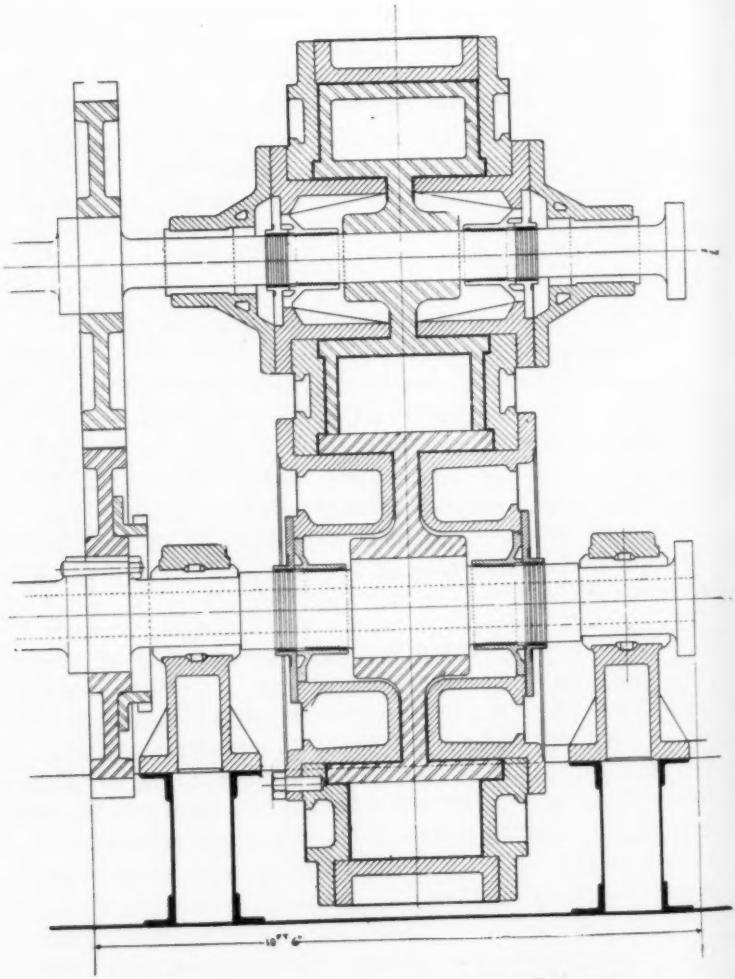


Fig. 3.—Herrick Balanced Rotary.—Section.

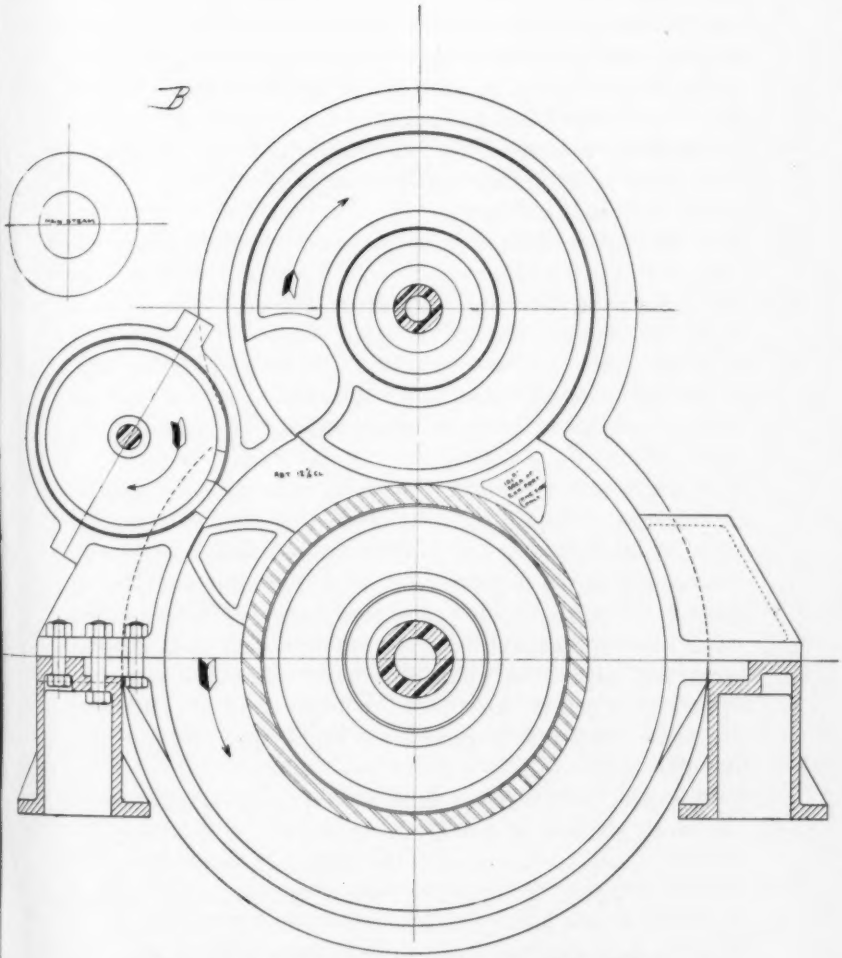


Fig. 4.—Herrick Balanced Rotary.—Elevation.

the essential vital features of the engine do not show details beyond what is necessary for a general idea of its construction and operation.

The engine structure comprises three moving parts enclosed within a casing having the general contour of the figure 8 and made to synchronize by gearing outside the casing. It must not be understood that gearing is used to transmit any portion of the propulsive power, but only that such a form of preserving synchronism is simplest and most positive.

The moving parts are:

1. The upper drum, or rotor abutment, provided with a recess in which the piston of the rotor "meshes," without contact, as it passes this drum at each revolution.

2. The circular valve on the left.

3. The rotor itself or lower drum with its fixed blade piston.

The valve admits steam to the cylinder chamber behind the piston blade shortly after the blade has left the recess in the upper drum.

In the elevation the moving parts are shown at a point where admission is to begin; after the required amount of steam has been admitted, the opening in the valve ceases to register with the opening in the cylinder casing and, the steam supply being cut off, the steam in the cylinder expands against the piston blade rotating the lower drum and the main shaft and the upper drum and valve in the direction of the arrows, exhausting after approximately three fourths of the revolution and just before the piston blade re-enters the recess in the upper drum to pass the abutment.

The casing, which forms the cylinder and encloses the drums or rotors, consists of a center body and two side plates. To the side plates, concentric with the upper and lower shafts, are fastened plugs which enter the annular spaces on each side of the central webs of the drum and between their hubs and annular rings formed at their circumference. These plugs are provided with labyrinth packing and bearings for their respective shafts. They are also provided with chambers on the periphery, or outer cylindrical surface, of their annular ex-

tensions, which enter the drums. Each of these chambers at the proper time communicating with the cylinder, radially opposite to itself and thus being provided with steam at the pressure in the cylinder at that point. This steam in the chambers acts as a pressure film and accomplishes the balancing of the radial loads mentioned above.

As the lower rotor turns the area of the rotor, exposed to the radial load of the high-pressure steam, increases and, in like manner, the balancing area between the plugs and the inside of the rotor increases also. Thus the entire load surface may be exactly counter-balanced at all times during the revolution of the piston.

The abutment cylinder is packed by a check piece which prevents steam from passing upward around the drum, while the line contact of the surface of this abutment where rolling on drum of the rotor (the direction of rotation being against the flow of leakage) minimizes loss in this direction.

The piston "packing" is of an "approximate contact" character. The upper face has a check or tongue piece which slides in a groove from the side and which is limited in its motion in axial plane by shoulders properly fitted and provided for. Centrifugal force tends to keep this piece out as far as its adjustment permits, but this, if desired, can be augmented by spiral springs in circular recesses beneath the packing strip.

The sides of the piston are packed somewhat similarly, but the strips are there necessarily held out by springs as described above, the centrifugal force not operating to their extension to the designed limit. These pieces extend radially the full length of the piston and cover the ends of the packing strip on upper or end face, this latter being to that degree shorter than the length of upper face.

In all this piston packing actual contact with the cylinder is intentionally avoided, yet without sacrificing seriously in leakage, but in our combination all leakage is used in the turbine at good economy. Even where there is no regain of use from the leaks it appears from the tests that the energy lost is fairly compensated for by the lack of that piston friction usual to reciprocating engines of equal powers.

ENDURANCE TESTS.

Of far more value than any theoretical deductions are the results of actual tests of this motor made under the supervision of Prof. Frederick L. Pryor, of Stevens Institute of Technology, and which show that economy, durability and reliability absolutely necessary and heretofore unrealized in any rotary engine.

True, these tests were made on a motor of small power (approximately 20 B.H.P.) but no developments were overlooked, nor phases left unconsidered, which would be necessary for making the data entirely applicable to a larger unit. Indeed any discussion regarding the possibility of enlarging the engine for the higher powers necessary in the combination as proposed, is entirely needless in view of the extreme simplicity of construction, for it would scarcely be possible to exercise sufficient carelessness in the work as to prevent increased economy due to the addition of the Rotary.

The Herrick Rotary under test was placed in the plant of the Degnon Contracting Company, New York City, and operated under actual service conditions, where it was required to furnish the lighting loads of the Degnon plant at such times as the lights were needed for their work from August 14th to December 21st, 1909.

Professor Pryor's report on these endurance tests is summarized in the following findings:

1. The engine ran 1,685 hours out of 2,664 hours.
2. It operated for 66 days of 17 hours each.
3. It operated for 18 days of 24 hours each.
4. It operated consecutively for 149 hours.
5. Only two stops were absolutely necessary due to troubles with the engine, and these troubles were not serious.
6. The total steam consumption of the engine was not increased due to the endurance run.
7. The decrease in horsepower and slight increase in steam per B.H.P. can be explained by alterations made in the bearings of the engine. (Note: a slight departure from accurate

alignment resulted from hasty work in changing size of ball bearings in course of this trial.)

8. The steam consumption of the engine per brake horse-power is exceedingly good for the size of the engine, and it is equivalent to the steam consumption in many (reciprocating) engines of equal power now on the market.

9. The space occupied, the weight of the engine and the economy of the engine, compared with other engines on the market, indicate this to be a commercial engine.

Never before has the rotary received any such justification for its existence, and when we remember that these tests were made with atmospheric exhaust and in a small engine we can consider that such deductions by one whose statements must ever be conservative mean much. In a previous test of this small-size motor by the same engineer the water rate of 44.2 pounds per B.H.P. was recorded with a vacuum of 18 inches in the exhaust line, and of course a lower rate is certain for higher vacuum. This compares so favorably with the water rates of our usual auxiliaries as to bring the Herrick Balanced Rotary into a far broader field than that of a unit of the main engines of marine machinery. Unlike the turbine, it can be throttled down as can the reciprocating engine, without great proportional increase in steam consumption per unit of power. For marine work it can be schemed so that all the machinery of the ship is of circular-motion type. All the auxiliaries being either rotary or turbine (dynamo drives), highest pressures and superheat can then be used directly in the auxiliaries and their exhaust utilized in the main turbines to greatest efficiency.

CONCLUSIONS.

We are certain to eventually abandon reciprocating motion in all marine machinery! This is neither a prophecy nor a clairvoyance, but merely the statement of a great mechanical desideratum awaiting the means for economical realization. Here apparently is the open door to this accomplishment and at

a greatly reduced first cost, cost of operation and attendants, amount of care and work in upkeep, and in simplicity of design.

That the scheme is worth immediate recognition and adoption on a scale large enough to be of value to the mechanical world is, in the opinion of the writer, unquestionable, and the precedence in inaugurating such a radical change will surely not rest long unclaimed.

TEST OF THORNYCROFT BOILERS, FUEL-OIL APPARATUS AND FIREROOM FANS ON TORPEDO-BOAT DESTROYER *ROE*.

By C. F. Bailey, Associate.

On April 14, 1910, tests were made at the works of the Newport News Shipbuilding and Drydock Company to note the working of the Thornycroft boilers, fuel-oil burning apparatus and the fireroom forced-draft fans installed on the torpedo-boat destroyer *Roe*.

The installation for the ship consists of four boilers of the Thornycroft type, built by the Newport News Shipbuilding and Drydock Company, two being installed in each fireroom. Each boiler contains 4,500 square feet of heating surface. The boilers are fitted for oil burning only.

The oil-burning apparatus in each fireroom includes twenty-two oil sprayers and two oil heaters of the Thornycroft type, built by the Newport News Shipbuilding and Drydock Company, each having 66.7 square feet of heating surface.

The steam pumps are of the Davidson make. In each fireroom there is one light-service oil pump of the vertical simplex type, size 6-inch steam cylinders, 7-inch oil cylinders, by 12-inch stroke, for pumping oil from the bunkers to the tanks abreast the boilers. There are two fuel-oil pressure pumps of the vertical duplex type, size 4½-inch steam cylinders, 2¾-inch oil cylinders, by 6-inch stroke, which draw from these tanks and discharge through the heaters to the burners. There are two main feed pumps installed in the engine room, of the vertical simplex type, with 14-inch steam cylinders, 10-inch water cylinders by 12-inch stroke. There is one combined air and circulating pump with 6-inch steam cylinder, 8-inch air cylinder, 8-inch water cylinder by 8-inch

stroke. There are several other pumps not used in the test which are not listed.

The auxiliary condenser contains 250 square feet of tube surface. It is fitted with ordinary $\frac{5}{8}$-inch tubes, No. 18 B.W.G. thick.

The forced-draft fans were furnished by the Terry Steam Turbine Co., and are of the Sirocco type built by the American Blower Company. Each fan is direct connected by a vertical shaft to a Terry steam turbine, as shown in Fig. 1. The fans are 30 inches diameter by $15\frac{1}{2}$ inches width of runner. The distributor is stationary and is of the form indicated in Fig. 2. The fans were designed to furnish 23,000 cubic feet of air per minute, against a pressure of 5 inches in the fireroom when running at 1,400 r.p.m. The bearings are oiled by a system of forced lubrication, which is self contained in the unit. Each fan engine is provided with an emergency trip valve to stop the engine in case of accident causing excessive speed.

For the purposes of the test the two boilers in the forward fireroom were used, together with the blowers and fuel-oil apparatus installed in this fireroom. Water for the boilers was taken from the city water mains and measured in tanks, as indicated in Figs. 3, 4 and 5.

As the tanks were filled they were emptied alternately into the feed tank, which was provided with a tight cover and air pipe. One of the main feed pumps in the engine room was utilized to draw from the feed tank and to discharge through the feed-water heater to the boilers.

Steam from the boilers was used for running the two forced-draft blowers, the main feed pump, the auxiliary air and circulating pump, the oil heater and the oil pumps. Live steam was also used for the feed-water heater. The balance of the steam was passed through a 4-inch valve, indicated in Fig. 4, in the engine room, and thence through 7-inch pipes overboard. These pipes were carried below the water, and in the top of each pipe was introduced a 2-inch salt-water spray with perforated nozzle to act as a partial condenser for the steam.

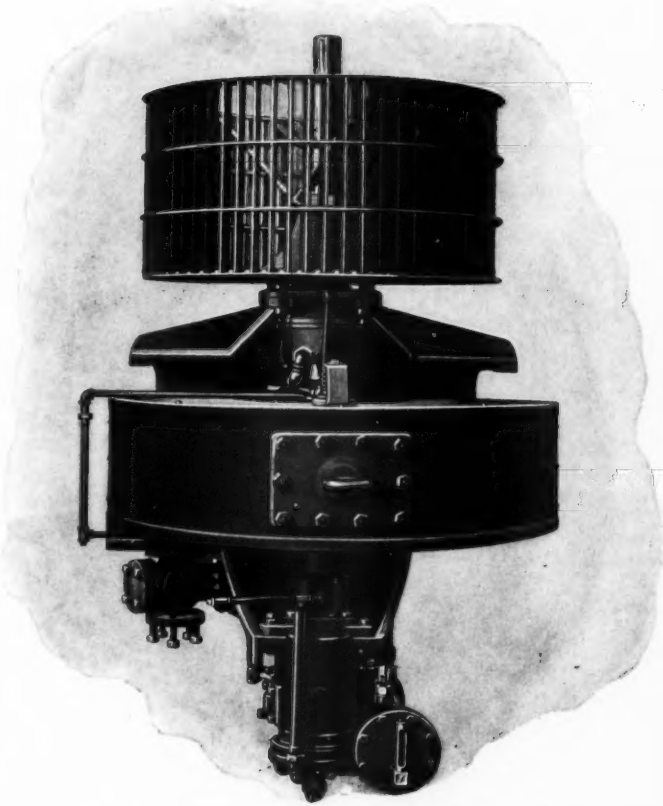


Fig. 1.—TERRY STEAM TURBINE AND SIROCCO FAN.

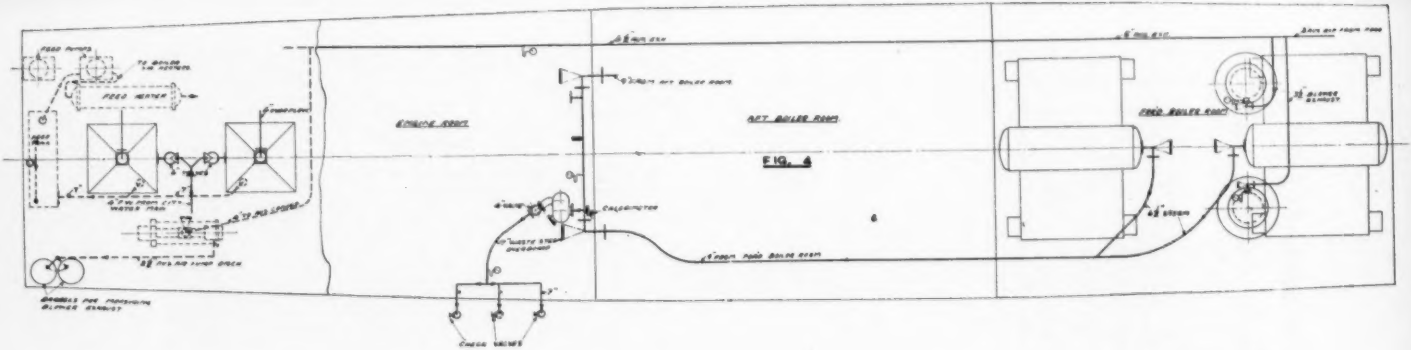


FIG. 4

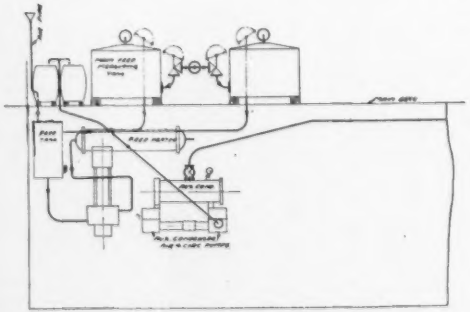


FIG. 3

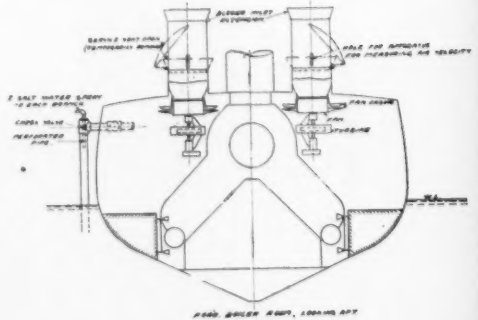
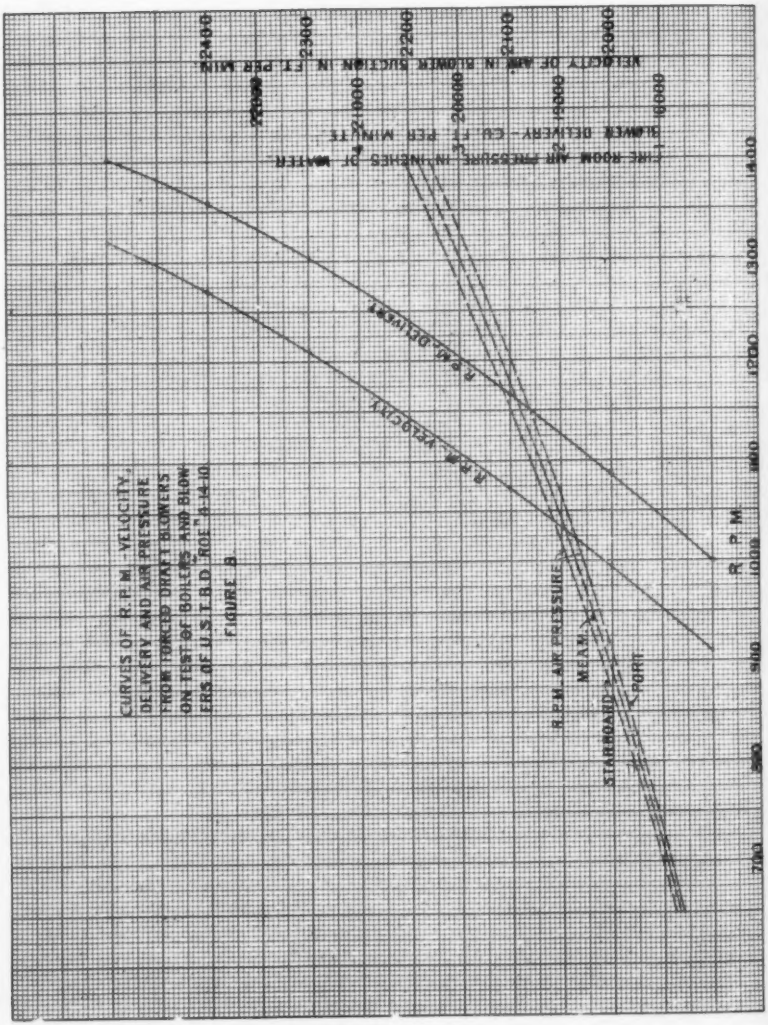


FIG. 2



CURVES OF R.P.M. - VELOCITY, DELIVERY AND AIR PRESSURE FROM FORCED DRAFT BLOWERS ON TEST OF BOILERS AND BLOWERS OF U.S.T.B.D. "ROE" 8-14-10.

FIGURE 8

Fig 8.

A small check valve was installed in each of these pipes to break the vacuum when shutting down. An additional 3-inch pipe was fitted overboard from the main steam, but at no time was it found necessary to use this pipe excepting for a short interval in test number four, and it is probable that the 4-inch valve would even then have cared for the total discharge. The 4-inch valve was regulated to maintain about 250 pounds at the boilers. A Barrus throttling calorimeter was placed in the main steam just before it entered the 4-inch valve in the engine room. The calorimeter was not lagged, and due to this and its distance from the boilers the quality of steam was probably better than indicated. The arrangement of disposing of the steam worked satisfactorily and with very little noise or vapor. A gauge installed in the discharge to the 7-inch pipes from the 4-inch valve showed no pressure during most of the test.

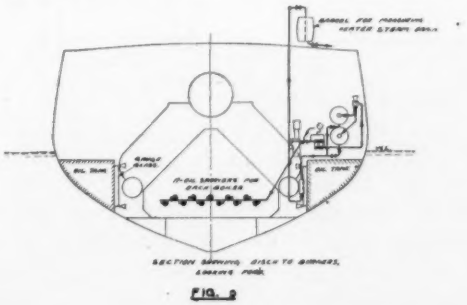


FIG. 2

The exhaust from the blowers was run to the auxiliary condenser in the engine room. The drain from this condenser was pumped to barrels on deck, where it was measured to determine the amount of steam used by the blowers. Live steam in the fireroom was taken to the oil heater, and the drain from this heater was taken through a drain collector provided with a gauge glass, and from thence was conducted into barrels on deck, care being taken to keep the height of the water in the drain collector such as to prevent steam from blowing through.

Extensions were made to each of the blower inlets, and air-measuring apparatus, consisting of Pitot tubes, was installed in connection with the starboard ventilator to determine the velocity of air down the ventilator, as indicated in Figs. 2 and 7. The amount of air handled by the port blower was estimated from a comparison of the revolutions of the port and starboard blowers. Each blower did its work well and ran without heating, and with very little vibration or noise, and from observations it was apparent that each fan was doing its proper amount of work.

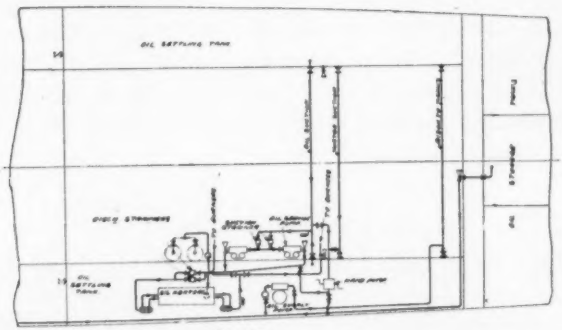


FIG. 10

The fuel oil was measured by noting the height of the oil in the gauge glasses on the oil tanks abreast the boilers. This was not thoroughly satisfactory in some of the tests, and the results are, therefore, not entirely consistent.

The fuel oil used was commercial oil, known as Illinois and Kansas crude.

Specific gravity, at 60 degrees F., . . .	0.88
Degrees Baumé, at 60 degrees F., . . .	30.0
Flash point, degrees F., ...	293.0
Calorific value by Mahler calorimeter, B.T.U., .	19,522.0

Tests were run at varying air pressures, and the results are tabulated in the summary of tests. The oil supply was regulated to suit the combustion by adjusting the burner open-

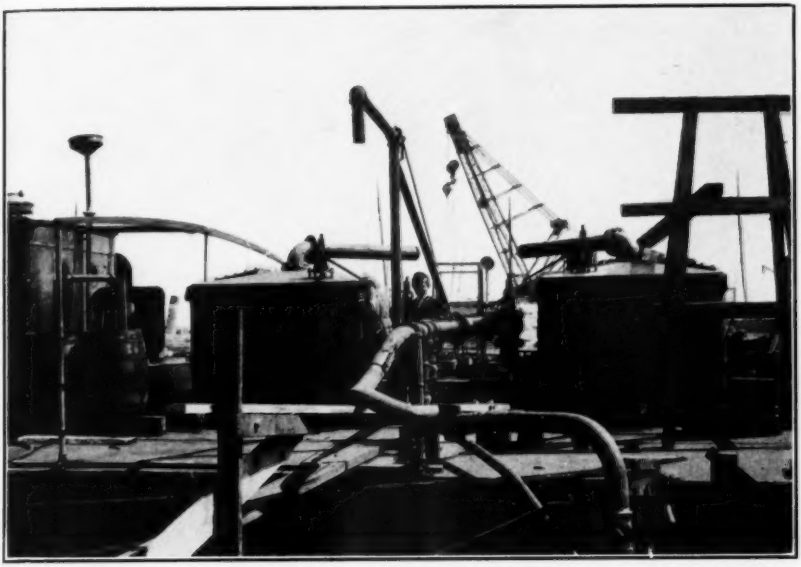


Fig. 5.—Feed-Measuring Tanks and Tank for Measuring Blower Exhaust.

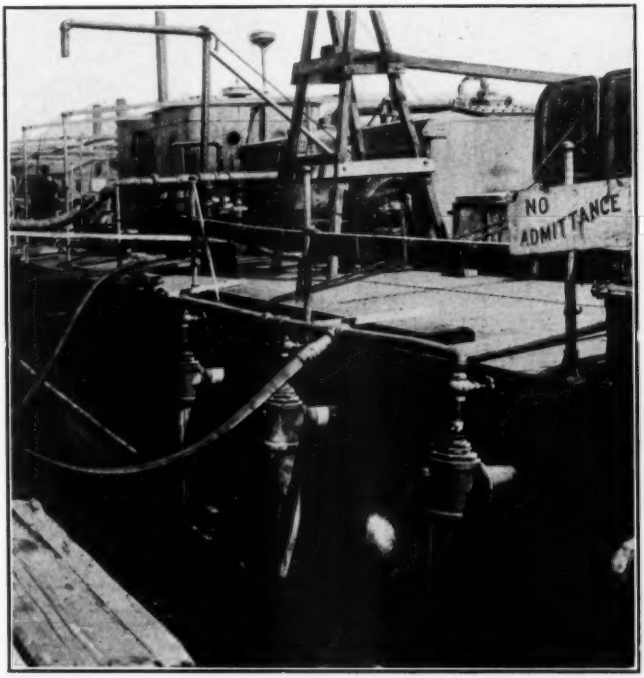


Fig. 6.—7-Inch Pipes Discharging Steam Overboard, with 2-Inch Spray Nozzle in Each.

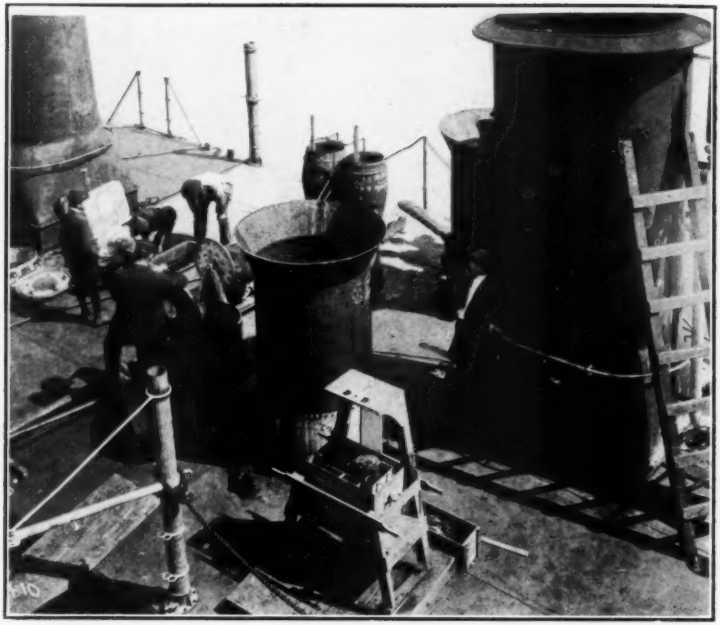


Fig. 7.—Extensions to Ventilators for Pitot Tubes, and Tanks for Measuring Oil-Heater Drain.

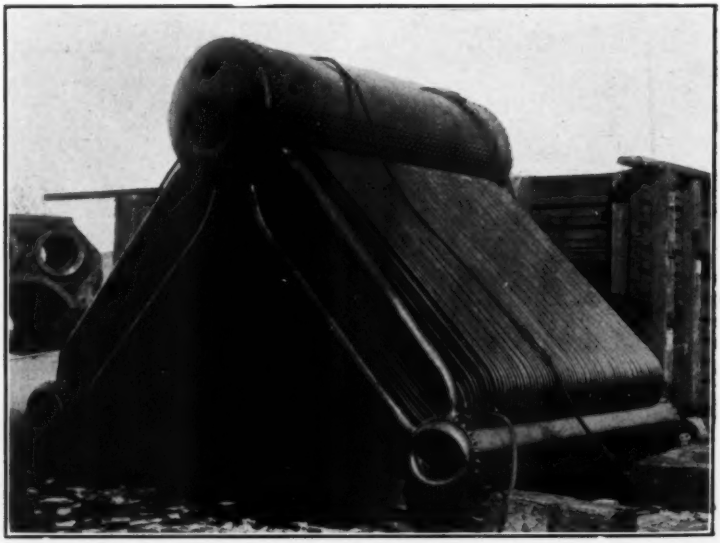


Fig. 11.—Thornycroft Boiler. Generating Tubes and Down Comers.

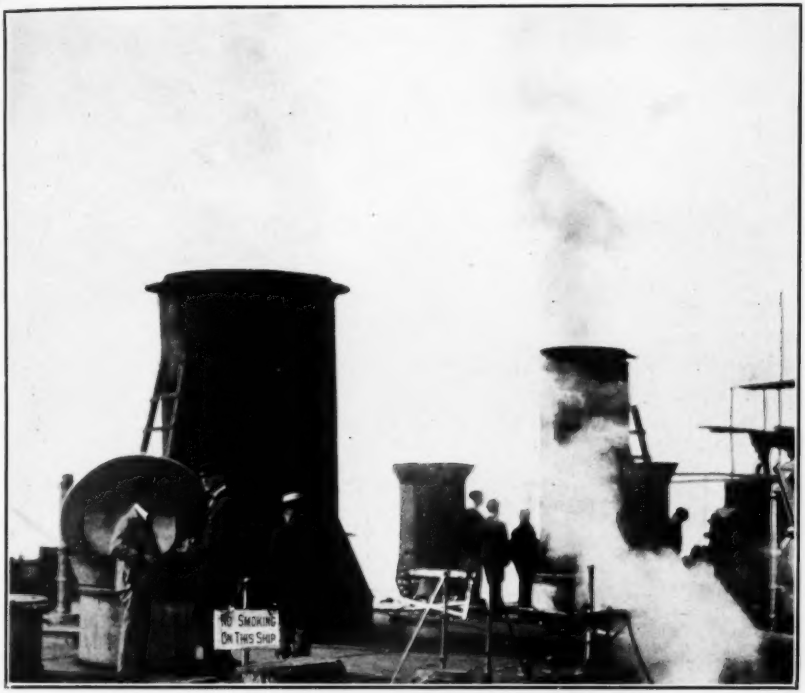


Fig. 12.—Test No. 1, Showing Smoke with 1.2-Inch Air Pressure.

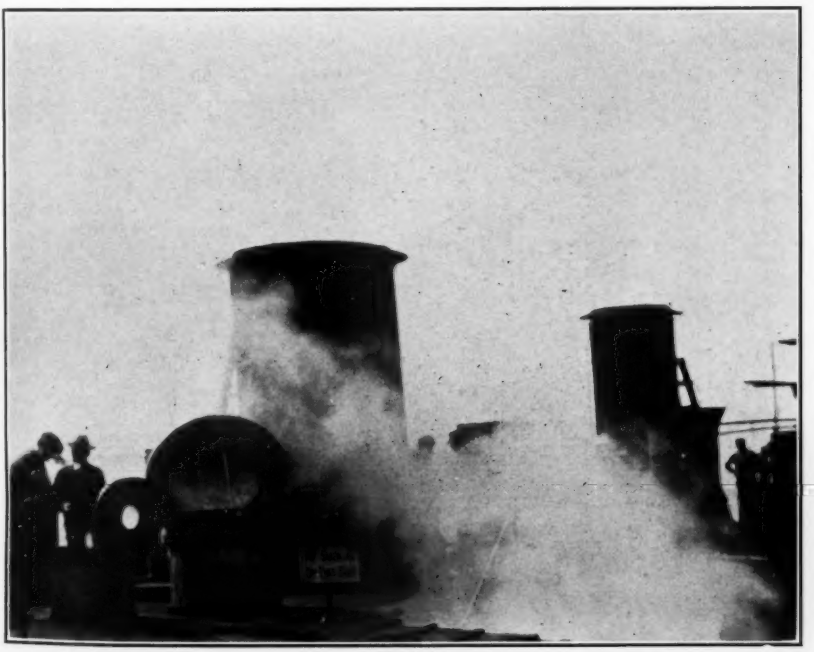


Fig. 13.—Test No. 2, Showing Smoke with 2.15-Inch Air Pressure.

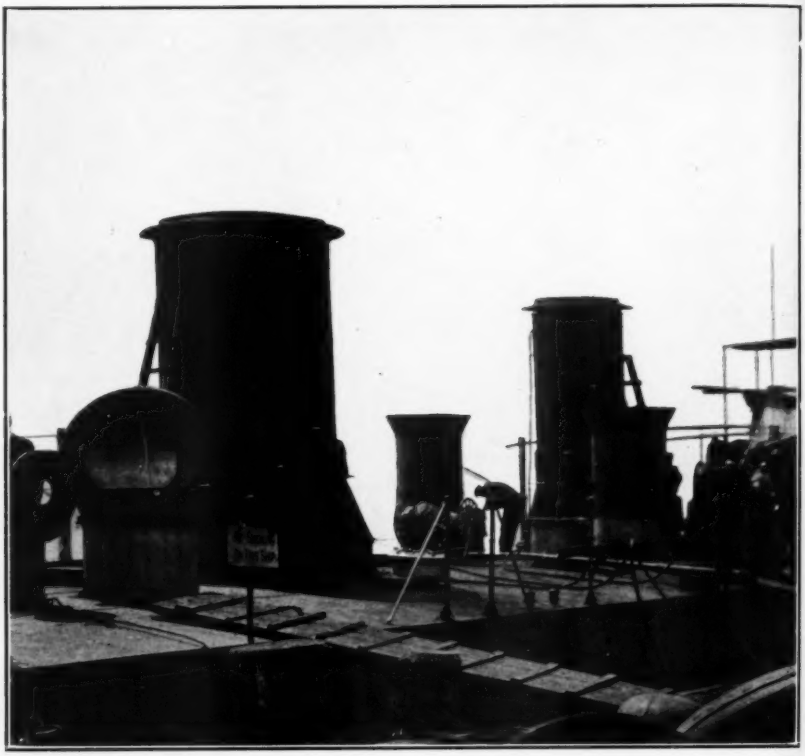


Fig. 14.—Test No. 3, Showing Smoke with 3-Inch Air Pressure.

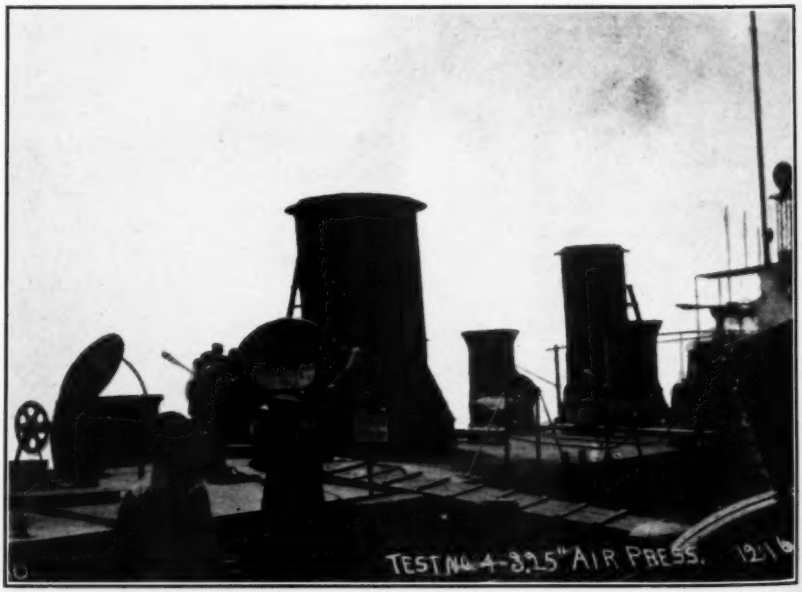


Fig. 15.—Test No. 4, Showing Smoke with 3¼-Inch Air Pressure.

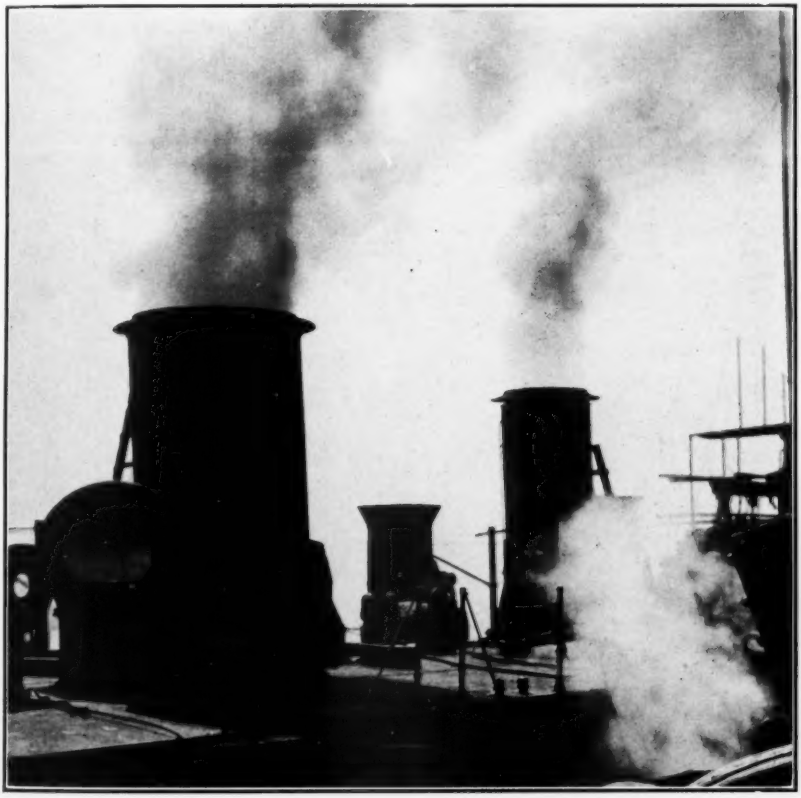


Fig. 16.—Test No. 5, Showing Smoke with 3.2-Inch Air Pressure, with Burners Opened Excessively.

ings to show a slight haze of smoke, to make sure that an excess of air was not being used. The burners could have been easily run to show no smoke at the funnel, but this was not done, owing to the excess of air which might thus be heated and wasted.

SUMMARY OF TESTS OF BOILERS AND BLOWERS, U. S. TORPEDO-BOAT DESTROYER *ROE*, APRIL 14, 1910.

Test No.	Time.	Duration, minutes.	Air press. in F. R., in ins. of water.	Temperature.					Boiler st'm gauge pressures.		Blowers.					
				Outside air.	Fire-room.		Stack.				Steam Pressure.		Exhaust Pressure.		R.p.m.	
					Above.	Below	Forw'd.	Middle.	For'd.	Aft.	P.	S.	P.	S.	P.	S.
1	8:31 9:41	70	1.2	61	87	94	...	...	242	246	70	69	−7″	−6.3″	805	767
2	10:05 11:05	60	2.15	62	83	91	820	810	241	246	142	148	3.0*	4.7	1,106	1,056
3	11:20 11:50	30	3.0	63	88	95	840	850	245	249	223	225	11.9	14.0	1,348	1,275
4	12:05 12:35	30	3.25	70	90	97	860	860	250	253	245	249	14.8	17.0	1,389	1,333
5	12:45 1:10	25	3.2	72	90	97	850	890	249	254	241	248	14.3	16.8	1,357	1,285

Test No.	Fuel-oil press.		Fuel-oil temp., degrees.		Steam pressure at oil heater.	R.p.m., oil pump.	Opening of burners in turns.	Fuel oil used, lbs. per hour.		Veloc. air, in feet per min., in starboard vent.	Cubic feet of air per minute to blowers.			Cubic feet air per pound oil	Temp. in feed tank.
	To filter.	From filter.	From sett'l'g tank.	From heater.				Total.	Per sq. ft. H. S.		Starboard.	Port.	Both.		
1	158	143	69	231	205	30.7	.35	5,240	.58	...	...	...	...	...	60
2	185	174	68	229	225	48.4	.6	6,780	.75	2,080	18,150	18,750	36,900	327	60
3	187	169	67	221	241	55.2	.95	...	...	2,410	21,040	22,250	43,290	...	59
4	177	169	70	213	250	59.0	1.1	8,400	.93	2,483	21,680	23,050	44,730	327	59
5	175	165	70	210	250	58.8	1.2	9,760	1.08	2,465	21,540	22,450	43,990	271	59

Test No.	Temperature feed water.	Per cent. dry steam.	R.p.m., main feed pump.	Aux. cond.		Pounds of water evaporated per hour.	Actual evaporation per		Factor of evaporation.	Evap. from and at 212 degs. per		Fuel-oil heater.		Per cent. water used by oil heater.
				R.p.m., pump.	Vacuum or press., lbs.		Square foot heating surf.	Pound oil.		Square foot heating surf.	Pound oil.	Pounds steam used per hr.	Lbs. oil heated per lb. steam.	
1	199	97.4	25 0	68	16.7	72,500	8 05	13.8	1.07	8.65	14.9	547	9 6	.75
2	199	97.4	23 8	74	21 6	76,700	8.53	11 3	1.07	9.16	12.1	688	9.9	.89
3	190	97.3	28 0	78	20 2	94,200	10 46	...	1.08	11.34	...	820	...	.87
4	187	97 6	28.4	78	19.1	102,700	11.4	12 2	1.09	12 41	13.3	853	9.9	.83
5	177	97 8	29.2	78	19 2	104,800	11 64	10.7	1.10	12.79	11.8	866	11 2	.83

Test No. 5 was run with an excess of smoke, the burners being 1.2 turns open instead of 1.1 turns open, as in test No. 4. This was done to note if a gain in evaporation could thus be obtained. As will be seen by the summary of tests, the total evaporation was slightly increased, but at the expense of an excess of oil. During the tests only one heater and one pressure oil pump were in service.

It is to be noted that with the vessel steaming ahead the power required of the fireroom fans will be reduced, and consequently the amount of air and pressure which can be obtained will be considerably increased, owing to the scoop effect of the ventilators, which will assist the fans.

The operation of the fans, boilers and oil-burning apparatus demonstrated that the outfit is capable of supplying ample steam for the power necessary to drive the vessel at the contract speed, and the ease of control and the satisfactory working of the apparatus was such as to indicate that the installation will prove economical and successful in service.

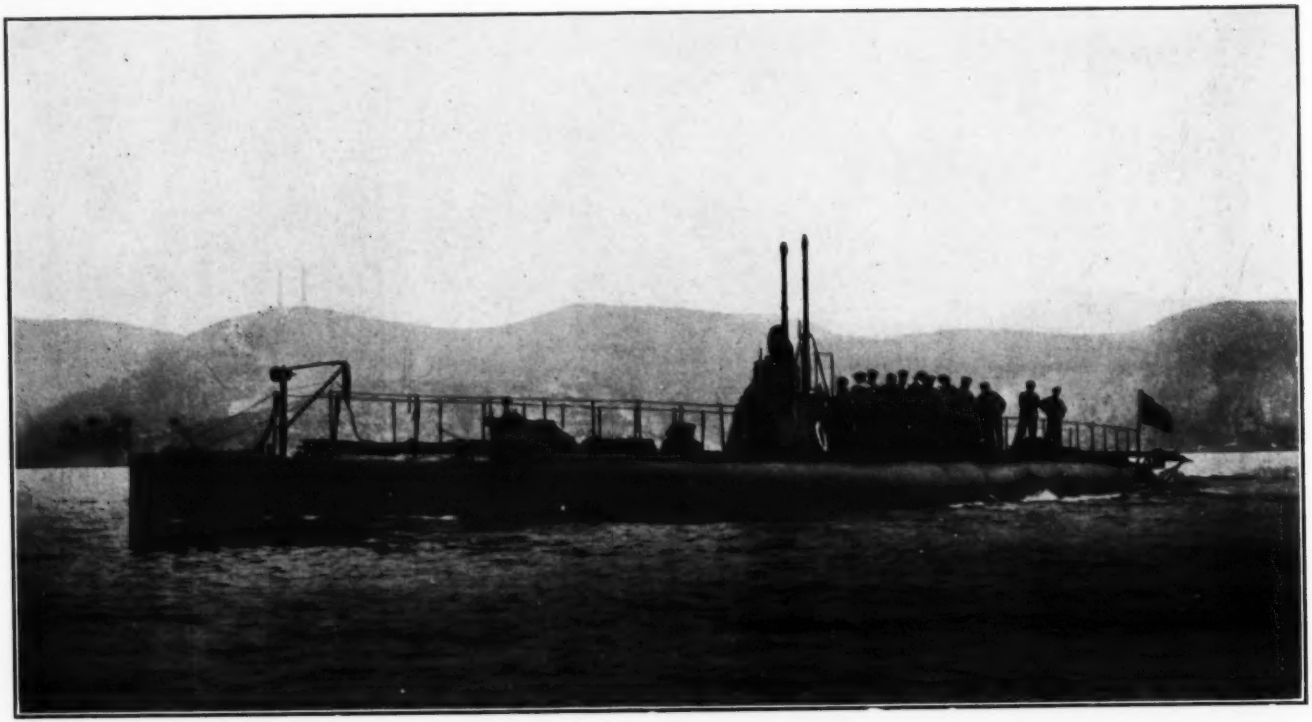


Plate I.—ITALIAN SUBMERSIBLE "FOCA."

THE SUBMERSIBLE.

By Robert G. Skerrett.

The United States naval authorities are now committed to the serious development of the under-water torpedo boat, and have opened the field to all competitors and to the best that the state of the art can produce. Our geographical conditions demand this encouragement to the designers of vessels of this sort. The greater reach of modern guns and the higher speeds of the heaviest fighting ships impose zones of activity which can not be met defensively by vessels of limited radii of action operating from harbor bases. These developments, combined with our coastal lines, call for an order of under-water torpedo boats which can cover considerable distances on the surface at relatively high speeds and which shall possess sea-keeping qualities of a marked character.

Speaking of the task set the designer of submarine vessels, it has been remarked, in a spirit of dissent, that he "Has had requirements imposed upon him arbitrarily, or in accordance with a general policy of naval tactics which has small regard for possibilities." This state of mind is unjustified. The under-water torpedo boat can properly be considered only in the light of a co-operative unit in any modern scheme of coast defense; and its value lies only in the extent to which it can work in unison with surface craft. Otherwise, it would be but little better than a mine field. The submarine torpedo boat fills one more gap in any line of defense, and when it is incapable of filling that gap, in accord with present-day requirements, then it has but little reason for being. It is but one cog wheel in a complex mechanism, and must work in

Robert G. Skerrett, Copyright, 1910.

harmony with the others. The development of naval tactics must, of necessity, impose the limits of usefulness of any fighting craft, and the builder of submarine boats suffers no more discrimination in this particular than the designer of any other type of naval vessel. The general trend abroad has been in the direction of developing under-water boats capable of taking their part in the wider spheres consequent upon the evolution of other ships of war, and to plead to the contrary is a confession of weakness or an admission that any particular type is fit only for merely local limitations.

A QUESTION OF TERMS.

Much confusion has grown out of the various efforts made to explain the difference between the terms "submarine" and "submersible" as applied to two distinct classes of under-water torpedo boats. Some writers have tried to read into the term "submersible" meanings more or less of a contradictory character; and this has been done in order to seem to justify the application of this denomination for the sake of lending more creditable dignity to the boats so described. In the language of the art, the "submersible" is commonly considered the acme of the science—hence the claim on the part of every builder that his boat is properly a submersible rather than a submarine.

Mr. Spear, in his paper published in the JOURNAL of last May, endeavored to clarify this terminology, keeping before him, however, the characteristics of the boats of his own design, which he put in the submersible class. Mr. Spear summed up the question in this manner:

"A fair conclusion from this is that the term 'submersible' as originally employed in France had no direct reference to the shape of the hull or to the amount of reserve buoyancy or the means adopted for submerged control, but was employed to convey the idea of a vessel designed to navigate on the surface propelled by heat engines in open communication with the atmosphere, but capable at will of submergence. The differences in form and buoyancy flowed from the supposed

requirements for surface navigation. In the course of events, however, these secondary considerations have been advanced to the dignity of primary and controlling elements. . . ."

Monsieur Laubeuf, the designer of the *Narval,* was the first to use the term "submersible" to distinguish his boat from those previously designed and known as submarines. Therefore it is fitting that we learn what Monsieur Laubeuf deemed to be essential in vessels of his own chosen denomination. This will clear up a good deal of confusion, even though, alas, it may bring discomfit to those builders of under-water craft that have called their boats "submersibles" without warrant in fact. Monsieur Laubeuf has given us this definite expression of his own conception of the two classes:*

"The real difference between the two types consists in that the submarine has, when maneuvering on the surface, but a small buoyancy (5 to 12 per cent. of its total *displacement* when submerged), and has a cigar-shaped or spindle-shaped body; while the submersible has great buoyancy under like circumstances (20 to 40 per cent. of its total displacement when submerged), and has the ordinary form of a sea boat. These essential characteristics make the submersible a seaworthy boat, but cause the submarine to make a poor showing on the surface. It also results that the submersible can be handled more easily, and hence show better speed on the surface than does the submarine; and, also, that it can run on the surface a much longer time than the latter, without having to close its hatches."

While Monsieur Laubeuf expressly disclaims any intention to signify by the term "submersible" a boat submerging on an even keel, yet he says:

"I have always adopted 'even keel' submergence for all my submersibles, of which I built 43. This is for reasons of safety of the boat and of facility of handling while descending."

Thus, we see that these features of form of hull and reserve

*"Journal U. S. Artillery," Nov.-Dec., 1907.

of buoyancy are not, as Mr. Spear puts it, "secondary considerations," but they are the vital essentials wherein the prime distinction lies between the two opposing orders of underwater craft. If the use of steam had been the impelling condition calling for great reserve of buoyancy and ship-shape form of hull, why have these characteristics been retained in all the French boats of the submersible class since the original conception of the *Narval,* even though the explosive engine has generally supplanted the use of steam?

The "submersible" was inspired by a policy of naval strategy which had for its aim the production of an under-water torpedo boat capable of leaving the French coast and attacking British naval craft in cross-Channel harbors. The "submarine" had too little freeboard to be a serviceable sea boat, and her limited radius of action made it quite impossible for her to reach the English coast, to make a successful attack, and to return safely to her home port. The "submersible," on the other hand, was designed by Monsieur Laubeuf to have good sea-keeping qualities and a considerable radius of action on the surface. The manner in which the offensive operation was to be carried out was as follows: "To leave a French port at dusk, cross the Channel under steam during the night, and to wait there throughout the day for the 'favorable occasion,' in the evening to relight the fires, and to return during the night under steam."

The use of steam was the stumbling block, for at that time the explosive engine was not deemed sufficiently reliable for surface propulsion. The fact that steam was used did not, in itself, constitute a distinctive feature of the submersible in the designer's mind, but was the only present source of reliable power then available for surface work. The steam plant of the *Narval* was incapable of giving the desired radius of action, and while the vessel proved an excellent sea boat she was not a fulfillment of the military requirements imposed. She blazed, however, the way for a distinct departure from the group of boats previously termed "submarines." Broadly considered, the submersible was the first effort to break away

from the limited mobility of the submarine, and in this fact lies our present interest in the distinction, because we, even more than France, require that our under-water boats should be vessels of considerable radius of action and capable of meeting the boisterous conditions imposed by the Atlantic and the Pacific during any season of the year.

TYPE LIMITATIONS.

Practical experience and model trials in experimental basins have plainly established the fact that surface and submerged navigation have their separate limitations. The form of hull which can be safely and speedily driven on the surface can not be propelled with the same propulsive economy under water. Again, the spindle-shaped hull offers less resistance when running submerged relatively than when being driven at its maximum speed on the surface; but it has an inherent tendency to bury its bow and to plunge when running either with reduced buoyancy on the surface or when running submerged and near the surface. In a considerable seaway, therefore, it is unwise, if not dangerous, to force the submarine at its highest surface speed either in its lightest trim or in any condition of reduced buoyancy. The loss of the *A-8* of the British navy laid a distressing emphasis upon the risk run in the latter condition.

For speed under water, the submarine is, therefore, an easier boat to propel than the submersible submerged; and the problem resolves itself into the broader question of that all-around military efficiency demanded by geographical circumstances. To this problem also must be added that important factor of a proper measure of habitability, so that the crew—even after days at sea—may be fit for their task when the hour of actual trial arrives. Too much stress can not be laid upon this feature of the question: the personnel of the boat should not be subjected to physical conditions that are any harder than those to be met in a surface craft of corresponding displacement.

Therefore we must make some sacrifice, and at present our

compromise forces us to surrender a measure of the submerged maximum speed. This need not, however, require us to seriously curtail the submerged endurance at a good speed for a four-hour period; this mobility and speed being the true practical value of the under-water vessel.

Neither ship-shaped form nor a large reserve of buoyancy alone will constitute a submersible within the meaning of Monsieur Laubeuf. The fact that a submarine of the usual spindle-shaped hull has its buoyancy augmented beyond the submarine limit described by Monsieur Laubeuf does not put the submarine in the submersible class, because the dynamic conditions peculiar to the cigar-form in motion are still present and hamper the surface operations of the boat. Again, to obtain this reserve of buoyancy by a superposed fairwater or superstructure, while adding in a measure to the surface capabilities of the submarine, discounts the submerged qualities. No submarine, so modified, upon equal displacement, has yet been developed which is the offensive military instrument that the submersible is in its best evolution.

The French have been accused of wasting much money in their scattered manner of building a varied flotilla of underwater boats, but the result has been an exceptionally wide experience, and a breadth of knowledge which has helped the rest of the world and will probably give the French invaluable direction for the future. It has been claimed in France, that in the matter of reserve buoyancy the true solution would probably be a compromise between 15 and 20 per cent. of the submerged displacement; the idea being that the sea-going qualities would then be ample while the economy of submerged propulsion would be more nearly akin to that of the submarine. The French naval authorities made a practical experiment on a full-sized scale to settle this moot point. The submersible *Sirène,* having normally a reserve of buoyancy on the surface of 27 per cent., had this buoyancy reduced to 15 per cent. by the admission of water ballast. Her sea-going qualities were found to be greatly impaired. The boat showed all of the surface disabilities of the submarine and the submerged limi-

tations of the submersible. In brief, she had none of the good qualities of either, but combined the bad ones of both. The reduced freeboard and the added displacement made the *Sirène* more sluggish in a seaway than normally; her deck was continually buried under succeeding seas, and she was hard to handle, even though her ship-shaped body did give her some advantage over the submarine under like weather conditions. The difference between the submerged and surface displacement of the *Octopus* is 35 tons, or a reserve of buoyancy in the light trim of 12.7 per cent. of her *submerged* displacement.

With the single exception of France, where the under-water boat has been developed by the Government alone, the present state of the art has been brought about mainly by commercial rivalry. The claim to superior qualities in any of the several commercial types can be justified only so far as it is a better military instrument. In his interesting paper, Mr. Spear has made certain deductions based upon *general averages,* and has contrasted the Holland submarine with a more numerous group of submersibles containing "lame ducks," old boats, and vessels of mediocre accomplishments. He naively says: "If individual cases are employed the comparison is really between the skill of the designers, and not between the general types." It is this contest between individual designers that gives us the true state of the art, and it is the development of the individual design that enables the prospective purchaser to properly evaluate the rival types submitted for his consideration.

FACTOR OF DISPLACEMENT.

During the submarine investigation in Congress in 1908, considerable time was devoted to discussing the proper commercial basis for evaluating the price per ton of under-water boats. There can be no doubt that the surface displacement, rather than the submerged displacement, is really the right one upon which to determine values. The heavier submerged displacement merely represents so much water admitted to the ballast tanks in order to bring the vessel to the desired con-

dition for facile handling under water. During that inquiry, Mr. Francis Tiffany Bowles, an extensive builder of Holland boats, gave pertinent testimony upon this point. He said:

"The light displacement of the boat, which is her actual weight in pounds, of course, is the measure of what the boat contains in materials and in structure, and therefore is the designer's capital upon which he works, and it is a vital element in the design and the outcome of the contract."

Mr. Lawrence Y. Spear, the designer of most of the recent Holland boats for the United States Navy has said:*

"Obviously the neglect of tonnage makes the comparison crude, as the displacement will affect the other qualities; and in addition substantial equality of tonnage is necessary to the assumed equality of features not included in the comparison, such as armament, habitability, etc."

Emphasis is purposely laid upon the foregoing quotation, because Mr. Spear has persistenly neglected this all-important element of displacement in making his comparative tables, in fact, has studiously set aside this factor when arriving at the deductions favorable to the boats of spindle-shaped form of hull and moderate reserves of buoyancy, *i. e.*, the submarines.

Fortunately, there are established formulae which give us a more reasonable means of accurately evaluating the qualities of speed, horsepower, endurance and economies of rival types, and we can do this without making apologies or resorting to the "crudeness" to which Mr. Spear confesses. As we are now building a submersible of the Laurenti type, it should be of interest to take some of Mr. Spear's tables—with corrected data—and see if we can not fill up the gaps and make a fairer and more instructive comparison. It is believed that by doing this we shall be better able to understand why our naval authorities have given encouragement to the Italian designer. Accepting, pro tem., the *Octopus* as typical of the best development of the submarine upon a surface displacement of 239 tons, let us take the Swedish submersible *Hvalen*

*"Proc. Soc. Naval Architects and Marine Engineers," Nov., 1906.

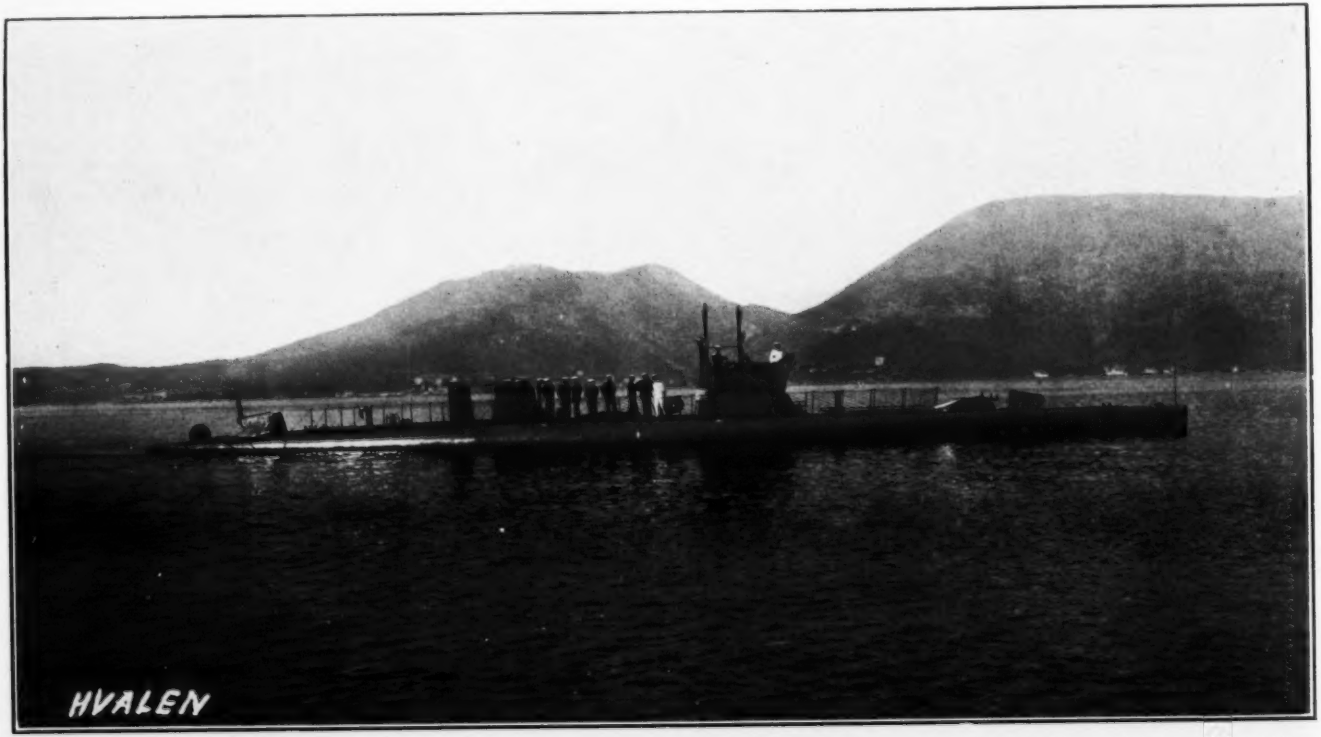


Plate II.—The Swedish Submersible "Hvalen" an Improved "Foca." The Submerging Rudders, Forward and Aft, are Arranged to Turn Up and Out of the Way When Running on the Surface.

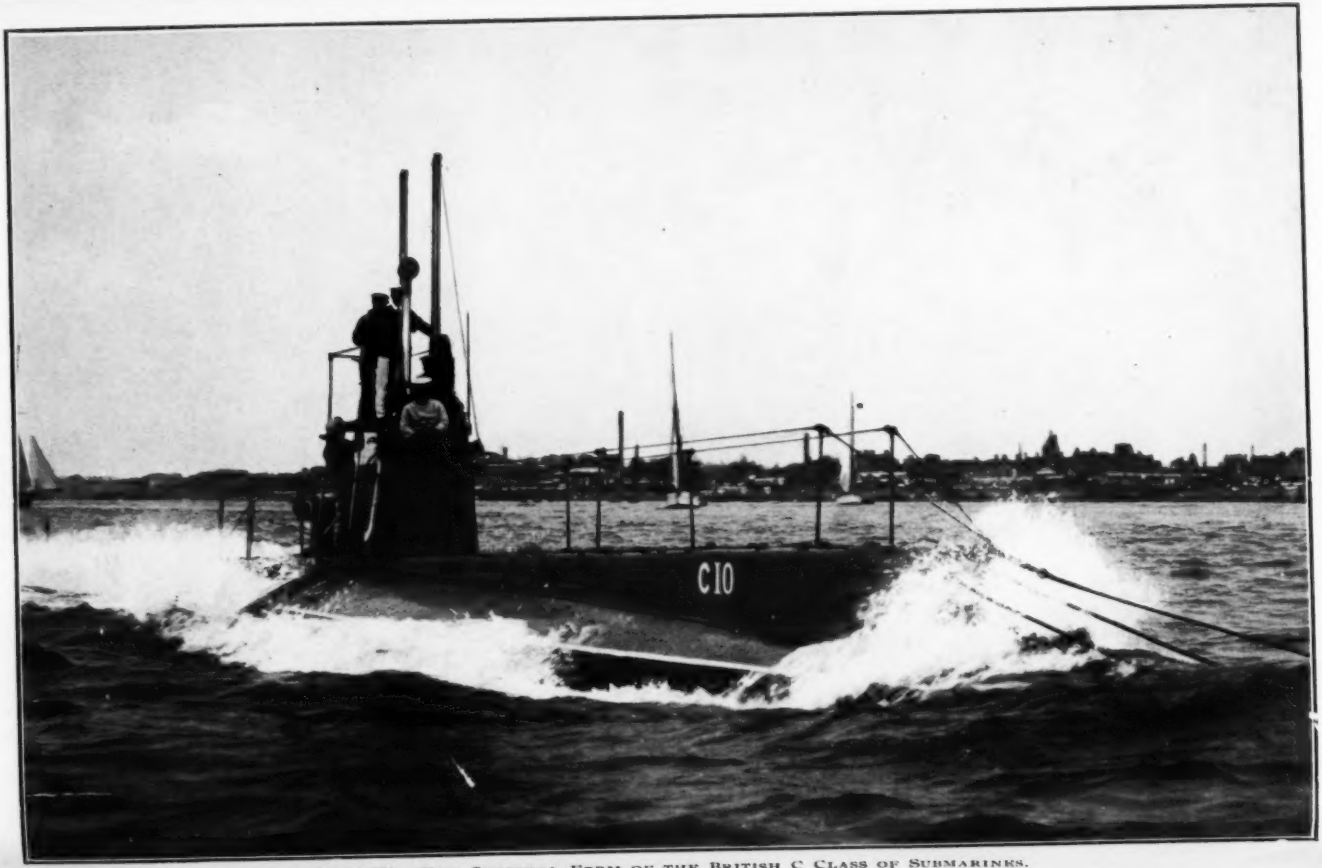


Plate IV.—The Original Form of the British C Class of Submarines.

—built by the Fiat-San Giorgio of Spezia—of a surface displacement of 186 tons, and see how the vessels compare just as they stand. The comparative figures are reliably as follows:

Name.	Surf. disp., tons.	Sub. disp., tons.	Reserve buoyancy, per cent.	Max. surf. speed.	Max. sub. speed.	Rad. of act. surface.		Rad. of act. submerged.	
						Speed.	Knots.	Speed.	Knots.
Octopus.....	239	274	14.60	11.0	9.93	11.00 9.35 7.25	445 680 1,120	9.93 8.00	... 32
Hvalen.....	186	230	23.65	15.2	7.00	15.00 11.00 10.00	470 975 1,500	7.00	21

Reserve buoyancy given in percentage of *surface displacement*.

A little analysis of the foregoing figures will enable us to make some instructive deductions. The surface displacement of the *Hvalen* is only 77.8 per cent. of that of the *Octopus*. Even upon this lesser displacement, the *Hvalen* retains a reserve of buoyancy of 44 tons as against the 35 tons of reserve buoyancy of the *Octopus*, the larger boat. This 35 tons of reserve buoyancy is the maximum margin of safety possessed by the *Octopus*, while the 44 tons reserve buoyancy of the *Hvalen* is the margin of safety which the latter retains *when her superstructure is open to the sea*. With her superstructure sealed by closing her freeing ports, the *Hvalen's* reserve of buoyancy can be further increased until it amounts to nearly 60 per cent. of her surface displacement. Any seafaring man will appreciate the significance of this. Proof of what this reserve of buoyancy means in sea keeping and speed making under service conditions was fully demonstrated during the *Hvalen's* remarkable run from Spezia to Stockholm, a condition precedent to the craft's final acceptance by the Swedish authorities. The boat had no escort, and made her runs and stops as indicated upon the small chart of her course. The

British officers at Portsmouth and the German naval men at Kiel were outspoken in their admiration of the boat's performance; frankly admitting that their own under-water boats were incapable of achieving such results, especially upon so modest a displacement.

The comparisons between the radii of action of the *Octopus* and the *Hvalen* at various surface speeds speak for themselves. The brake horsepower of the *Hvalen's* three engines—the boat has triple screws—is exactly double the brake horsepower of the two engines of the *Octopus*. The *Octopus* carries 30 per cent. more fuel than the *Hvalen* and should have a greater radius of action than the Swedish boat. The fact that she has not, means two things: her hull is harder to drive in surface trim and her engines are nothing like so efficient and economical as the explosive motors of Fiat design. The fact to be remembered in analyzing these endurance figures, is that the *Hvalen* while going at 11 knots—4.2 knots less than her maximum but equal to the *Octopus'* best—is able to substantially double the radius of action of the submarine, *and yet holds in reserve a supply of fuel which will enable her to recharge her batteries.* We must remember that the ultimate endurance of an under-water boat—either on the surface or submerged—resolves itself into the question of fuel capacity. In this case the results obtained are due both to the skilful design of the hull and likewise to the superior development of the explosive engine.

The *Hvalen* was designed for a submerged speed of 7 knots and to have an endurance at that speed of only two hours. As a matter of fact, her batteries actually yielded an endurance of three hours at the seven-knot speed. This result, of course, is largely due to the efficient character of her accumulators and electrical installation generally. This point and the factor of fuel supply, together with engine economy, must be given their true weights in any effort to predetermine, even approximately, the possibilities of rival types. Mr. Spear endeavored to handle this question mathematically by means of formulae of his own devising. He neglected the fuel factor and the variable

possibilities upon given weights of batteries and engines. As we have seen by the figures of the *Hvalen*, this neglect must vitiate the value of conclusions drawn by Mr. Spear's method. In addition to this, he assumed that the horsepower for surface propulsion at maximum speeds varied as the fourth power of the speed. This higher power factor is probably fully justified in determining the motive force needed to drive the spindle-shaped submarine at maximum velocity on the surface; but actual performances and extensive model tests of ship-shaped hulls of good form show that the power for these boats varies as the cube of the speed. The same is found to be the case for certain ship-shaped forms when driven submerged at their reasonable maximum speeds.

Now, let us see what can be done with a Laurenti boat having the same *submerged* displacement as that of the *Octopus* but with the characteristic range of reserve buoyancy in light trim peculiar to the Italian type. This will give us data by which we can make a closer comparison between the *Octopus* and a Laurenti submersible of more nearly similar surface displacement.* These figures will show us what a difference in the possibilities of an under-water boat only a few tons make.

Name.	Surf. disp., tons.	Sub. disp., tons.	Reserve buoyancy, per cent.	Max. surf. speed.	Max. sub. speed.	Rad. of act. surface.		Rad. of act. submerged.	
						Speed.	Knots.	Speed.	Knots.
Octopus.....	239	274	14.60	11.00	9.93	11.00 9.35 8.00	445 680 1,050	9.93 8.00	? 32
Laurenti.....	216	274	26.80	12.00	9.00	12.00 10.00	1,400 2,200	9.00 8.00 7.00	9 18 28

This Laurenti boat could carry three torpedo tubes and a supply of six torpedoes, as against two torpedo tubes and four

* Identical surface displacements remaining, however, the true basis for a just comparison.

torpedoes in the *Octopus*. The hull would be strong enough to withstand safely a submergence to a depth of 150 feet. The figures showing the radii of action speak for themselves. Let us reduce these figures to percentages of efficiency, as Mr. Spear has done, giving to the figures of the *Octopus* the unit value.

	Octopus.	Value.	*Laurenti.*	Value.
Reserve buoyancy, surface, normal.....	14.60 p.c.	100	26.80 p.c.	183.5
Reserve buoyancy, surface, maximum...	14.60 p.c.	100	52.00 p.c.	356.0
Radius of action, surface, full speed.....	$\frac{11.00}{445}$	100	$\frac{12.00}{1,400}$	314.0
Cruising speed.....	$\frac{9.35}{680}$	100	$\frac{10.00}{2,200}$	*320.0
Radius of action, submerged, 4-hour rate.	$\frac{8.00}{32}$	100	$\frac{7.00}{28}$	87.5
Torpedo tubes.....	2	100	3	150
Torpedoes.....	4	100	6	150
Submergence depth, feet.....	200	100	150	75.0
Drop keel.....	0	0	1	100

Mr. Spear says: "A complete and accurate comparison of military value would require that all vessels have the same displacement, the same strength to resist submergence pressures, and be substantially equal in seaworthiness, habitability, armament and durability."

The preceding table takes Mr. Spear at his word in every particular save the features of total surface displacement and strength of hull equal to hydrostatic pressure at a depth of 200 feet.

Mr. Spear tells us in his paper in the JOURNAL of last May that:

"The speeds (particularly the surface speed) achieved with any given type will depend largely upon the displacement, and, as the cost is practically proportionate to the displacement, the latter should be held to the smallest limit compatible with the service required, in order that the number of units may be increased for a given expenditure, or the expenditure minimized for a given number of units."

It was this very idea of obtaining the greatest number of

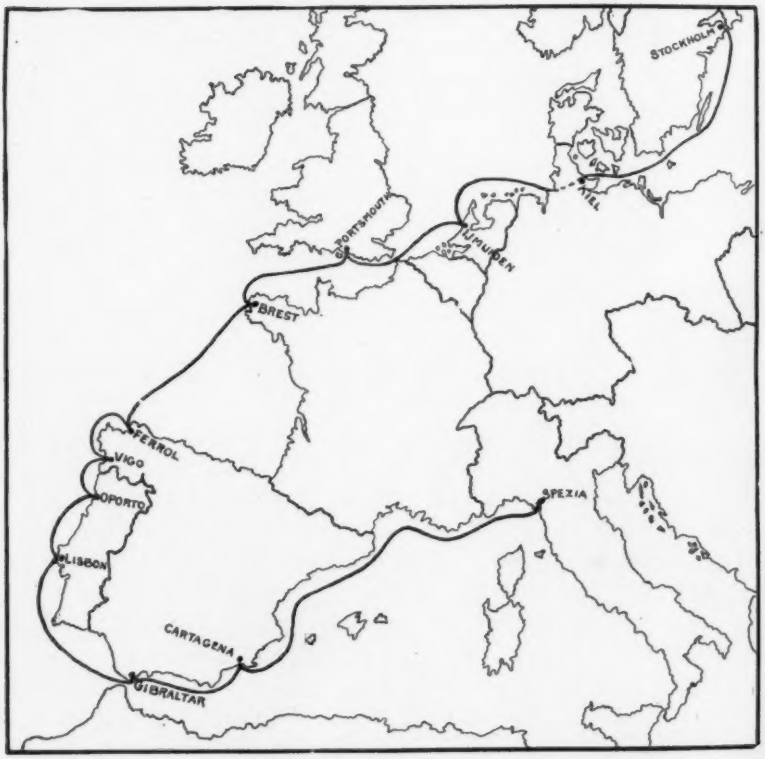
*This value is really greater, because the cruising speed of the Laurenti boat is higher than that of the *Octopus*.

units of defense upon a prescribed expenditure that influenced Engineer Laurenti in the original design and development of his submersibles. The results obtained by the Laurenti boats are a flat contradiction to Mr. Spear's dictum that surface speeds particularly depend largely upon displacement. The *Hvalen* and the *Foca* are boats of only 186 tons surface displacement; the first made 15.2 knots and the second 15 knots. The *Octopus,* having a surface displacement of 239 tons, has made but 11 knots light under the most favorable conditions, while the *Hvalen* has maintained a sea speed of 12 knots an hour when contending with heavy weather and running under only two of her three screws. Upon a given sum, more effective and speedier Laurenti boats can be built than the same sum will buy boats of the *Octopus* type of eleven knots.

SEAWORTHINESS.

It is quite impossible to reduce to mathematical values the highly-important elements of seaworthiness and habitability. These can be judged only by actual records of sustained performances. None of the submarines in the United States Navy has made an uninterrupted run of more than 200 miles, and the best performances have been made under favorable weather conditions and then attended by a mother-ship. When any of these boats have been subjected to nasty weather they have been pretty hard hit, and their crews have been badly used up. The *Hvalen's* longest sustained run was from Spezia to Cartagena, during her unconvoyed trip from Italy to Stockholm. By the usual trade route the Spezia-Cartagena run is a matter of 641 nautical miles. The *Hvalen,* however, followed the coast line and thus made the distance 790 miles, which she covered in the remarkable time of 72 hours; arriving at Cartagena in good shape and with her crew in excellent condition. During her total run of nearly 4,000 miles to Stockholm, the *Hvalen* encountered considerable heavy weather which fully tested her sea-keeping qualities, and her behavior under all circumstances aroused the admiration of her Swedish

commander. Throughout the whole trip the boat suffered but one mishap—a small and brief disablement of her steering gear, which was speedily repaired by the men on board. This mishap occurred during foggy weather and in the North Sea. As her commander was not then sure of his position, and the sea being heavy, the skipper had the *Hvalen* towed into near-by Ijmuiden, Holland, by a Dutch steam trawler.



TRIP OF THE "HVALEN."

In July of 1908 a flotilla of British submarines of the *B* and *C* classes—vessels of 800 tons submerged displacement, designed, so it is said, by Rear Admiral Reginald Bacon—made a non-stop run of 390 miles from Dover to the Firth of

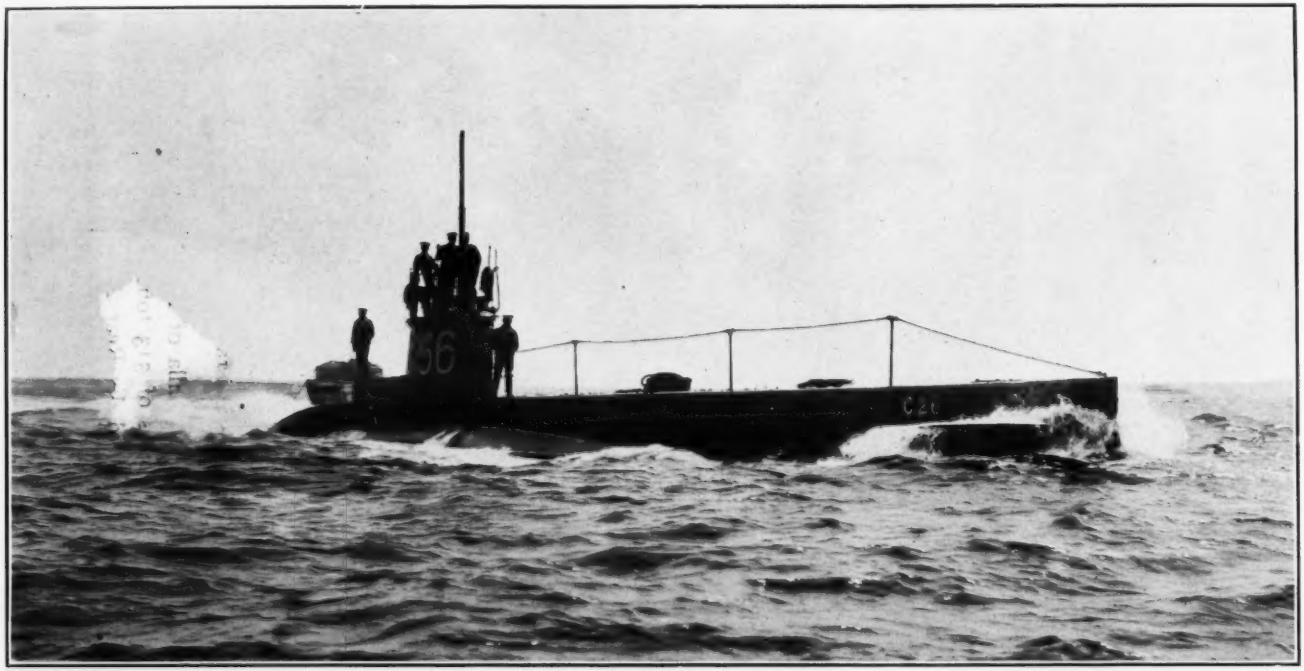


Plate V.

THE IMPROVED C CLASS OF BRITISH SUBMARINES. SHOWING A MORE EXTENSIVE SUPERSTRUCTURE AND THE ADDITION OF BOW RUDDERS: THE LATTER TO FACILITATE SUBMERGING AND SUBMERGED CONTROL WITH SMALLER CHANGES OF HORIZONTAL TRIM.

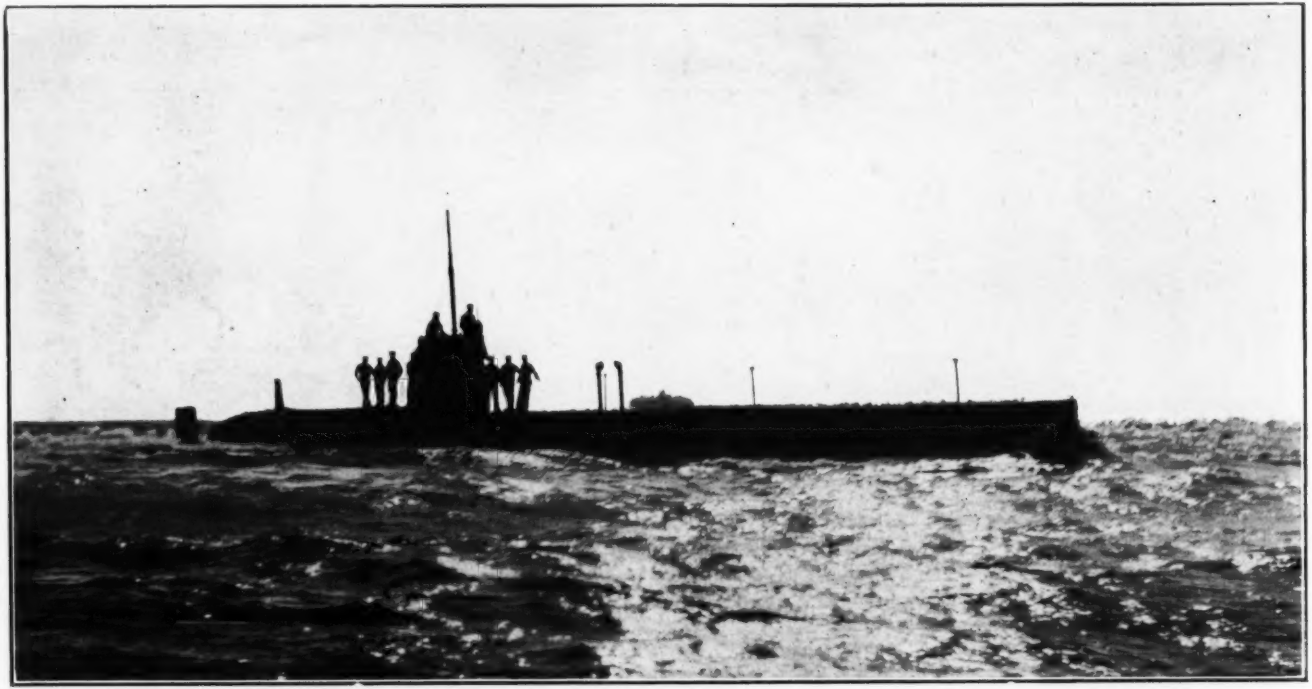


Plate VI.

"D 1," THE LATEST DEVELOPMENT OF THE BRITISH SUBMARINE. THE BOW RUDDERS ARE HIDDEN BY THE BOW WAVE. THE SUPERSTRUCTURE IS INCREASED IN HEIGHT AND EXTENT.

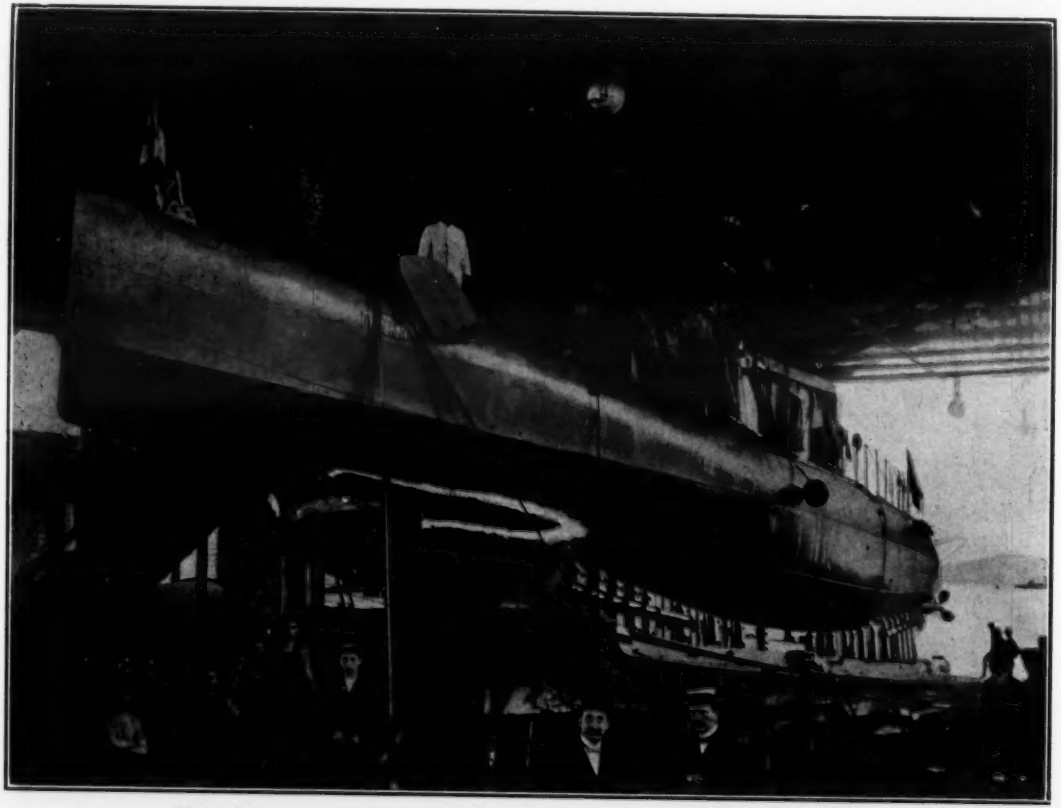


Plate VII.—Danish Submersible "Dykkeren" (with Form of Hull).

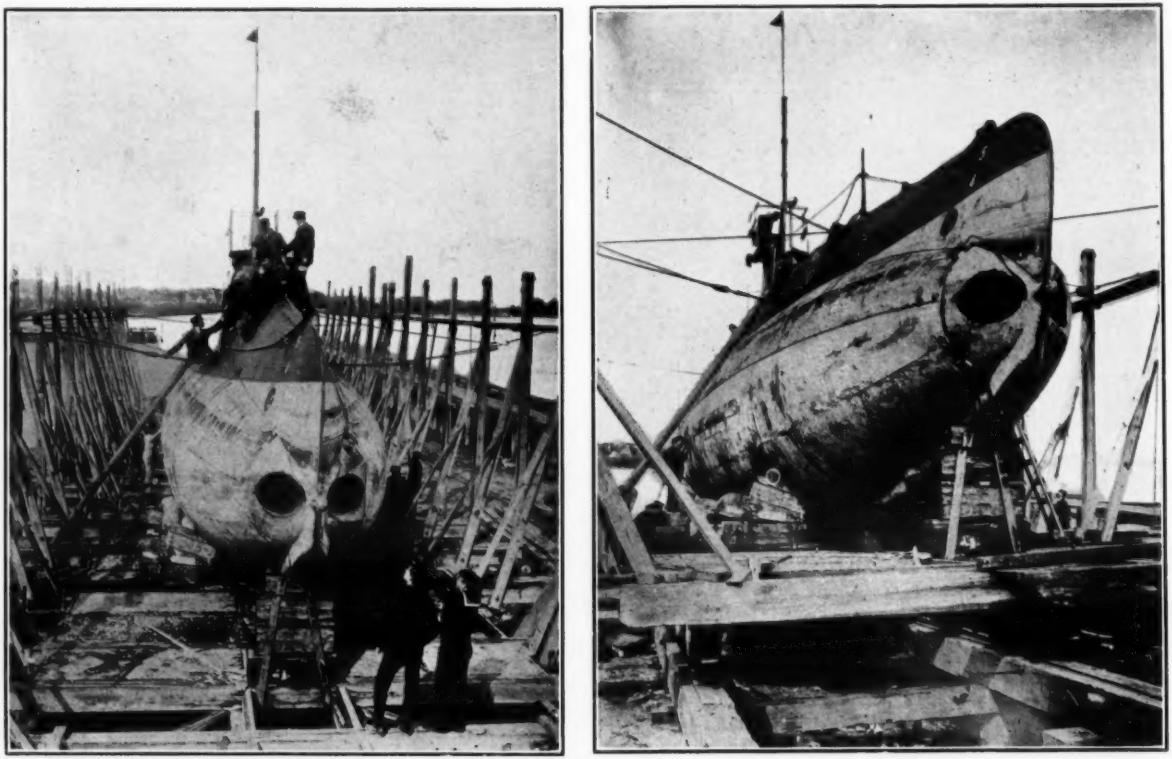


Plate VIII.—The U. S. Submarine "Octopus," Showing Plainly the Extension of the True Spindle Hull Modified Only by the Addition of a Partial Superstructure. Note Position of Bow Tubes and Their Exposure to Collision.

Forth in 40 hours. This was considered so remarkable by the naval authorities that a congratulatory letter was issued by the Lords of the Admiralty to the Commander in Chief under date of August 6, following. Mr. Spear has quoted this letter but overlooked the signature. This letter of praise might mean more or less if we knew its author. It is perfectly plain, however, that the boats of the *B* and *C* classes do not answer the demands of seaworthiness, because the *D-1,* a vessel of 500 tons submerged displacement, is a direct effort to obtain a submarine of better sea-going qualities. The height and extent of the superstructure show the direction in which the Admiralty has found it necessary to improve in order to make better and safer sea boats of its submarines. Compare this displacement with the corresponding displacement of the *Hvalen!* These changes are all in the direction of getting away from the true spindle form of hull with its attendant disabilities; and, with the single exception of superior strength submerged for a given weight of material, the retention of the circular cross section is thus admittedly undesirable.

Regarding steadiness in a seaway, it has been said of the spindle-formed submarine that "the rolling is nearly eliminated and is replaced by a curious lateral translation of the entire ship. This is especially true where the superstructure is non-watertight, and waves, coming aboard on one beam pass partly through the scuppers in the superstructure and disappear on the other beam."* The submarines of the *Adder* class, as first completed, were not fitted with bilge keels, and they rolled between 28 and 30 degrees in a seaway, in fact rolled so heavily that they spilled the electrolyte out of their cells. The *Fulton,* a sister boat, before she was submitted to the Government for official examination, received bilge keels in order to lessen her tendency to roll. After being fitted with these bilge keels, she managed to roll a matter of 20 degrees in a seaway when the waves coming abeam ranged from six to ten feet in height. This does not sound very much like extreme steadiness, nor does it exactly coincide with the idea of that "curious lateral

*"The Design of Submarines," "Soc. N. A. and M. E.," 1909.

translation" referred to. All the U. S. submarines since built have received bilge keels!

The *Glauco* class of submersibles, the first of the Laurenti boats built for the Italian Government, were fitted with bilge keels to check their rolling in a heavy sea, but the subsequent sea service of those submersibles proved that the bilge keels were quite unnecessary. The later boats of Laurenti design —the *Foca, Hvalen* and *Dykkeren*—have not been fitted with bilge keels, and their steadiness in a seaway is far better than that of surface torpedo boats of corresponding displacements.

The latest submarines all show higher and more extensive non-watertight superstructures. While these superstructures have fairly numerous freeing-ports, still the area and height of this surface exposed to a sea coming abeam is far greater relatively than the area of the freeing-ports through which the oncoming waves might pass unobstructedly. Therefore, if these boats, by reason of their lesser buoyancy, can not rise to a wave as a submersible would, or roll easily instead of being disposed to move laterally in the "curious" manner already described, they would be badly jarred by the impact of every good-sized wave coming aboard more or less on the beam. One can conceive that a superstructure built after the fashion of a sieve would give quite the action mentioned in the preceding quotation, but a superstructure of this description would hardly meet the practical requirements.

Concluding this question of the relative sea-keeping qualities of the Laurenti submersibles and boats kindred to the submarine *Octopus,* it may be remarked that last year, the Italian flotilla—boats of only 150 tons surface displacement—were kept at sea continuously for a period of more than a month without communication with the land. They were accompanied by their supply and mother-ship, the *Lombardia,* and the Italian authorities were reasonably very much gratified by the performance of the submersibles and the condition in which the crews were maintained throughout this period. Neither submersibles nor submarines in any other country have yet established a record of this sort.

FORM OF HULL.

The early idea of the submarine was a vessel capable of simulating the actions of a fish, and in order to give to the boat the best lines for submerged propulsion a fish-like or spindle-shaped form of body was commonly considered necessary. Apart from this, in order to increase the margin of safety against sudden or unintentionally deep submergence, a circular cross section was adopted, because it assured the greatest strength for a given weight of material. The prime aim was to build a craft which should have the greater part of its body well submerged, and even in its lightest condition only a very small part of its body should be above water—the supposition being that the vessel's usefulness would be measured by the length of time it could run completely submerged. It is not necessary to trace the development of under-water boats through all of the succeeding stages. All that need be known is that the submarine of very limited reserve buoyancy soon proved to be unequal to the wider task demanded of it through the evolution of other fighting craft. The designers still commonly held tenaciously to the spindle-shaped hull and circular cross section for reasons of structural economy and maximum strength, and, too, because this form—that is, the true spindle—was easiest to propel under water. However, in order to make these boats more capable of running safely at higher speeds at the surface, they were fitted with light, non-watertight superstructures; but just so far as these superstructures were expanded to offset the surface disabilities of the spindle form, the submerged speeds for a given power were diminished, while, at best, the energy required for the higher surface speeds became notoriously greater than the power needed to drive a ship-shaped body of corresponding displacement. Besides, the fact became more apparent that the under-water boat would have to operate upon the surface by far the larger part of the time, and the limitations of the spindle hull became in this way more and more accentuated. Model experiments and the undesirable—yes, even hazardous performances

of some of these submarines—showed that the spindle-shaped body, modified or otherwise, was not the best solution of the problem.* As the "submarine" stands today, it represents an intermediate growth between the true spindle hull and the ship-shaped body, and does not possess the best qualities of either. The excuse for the spindle hull remains justified only so far as those boats are concerned which are designed to dive in submerging rather than to go down more nearly on an even keel. This operation involves certain risks—especially in waters of moderate depth—which make it desirable to retain the circular section and the spindle form in the name of needed security. The author has previously pointed out the limitations of the diving boat, and the evolution of the *B* and *C* classes of British submarines bear evidence to the Admiralty's appreciation of these limitations; the increase of the superstructures and the adoption of hydroplanes and bow rudders to restrict the change of trim when submerging or change of vertical position while running under water are all efforts to break away from the disabilities of the spindle-shaped and diving submarine. Every added foot of length in the case of a diving boat means an increase to the difficulty of her safe control in waters of moderate depths; and every increment of submerged speed adds accordingly to the hazard of reaching quickly a dangerous depth through inattention on the part of the diving rudderman.

The reason for the tendency of the spindle-shaped submarine to plunge of its own impulse has been variously explained, but some of these explanations have been involved and have generally overlooked the simple principle at the bottom of the whole problem. It is a matter of common knowledge in hydrostatics that, where water is flowing from a large pipe into a small one and then into a large one again, the maximum pressures are exerted on the walls of the larger pipes and a reduced pressure and greater speed of flow are present in the smaller conduit. (See Fig. No. 1.) This really is the key to the impulsive diving tendency of the spindle-shaped body.

* See "Journal U. S. Artillery," Nov.-Dec., 1906.

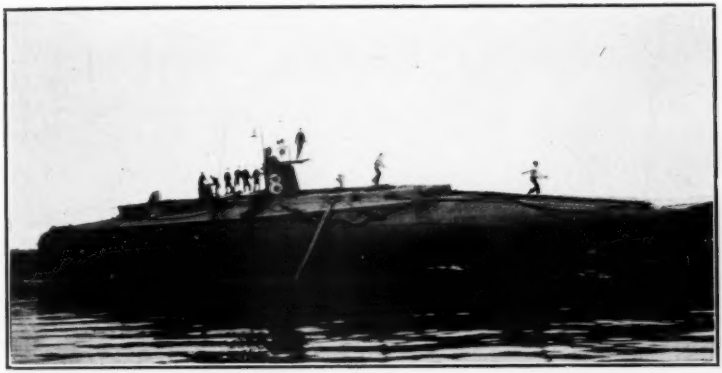


Plate IX.—The British Submarine "B 8," Ashore. Showing the True Spindle Hull and the Nature and Extent of the Superstructure.

Plate X.—The U. S. Submarine, "Narwhal," Showing the So-called "Modified Spindle," which is Modified Mainly Only by the Extent and Height of the Superstructure.

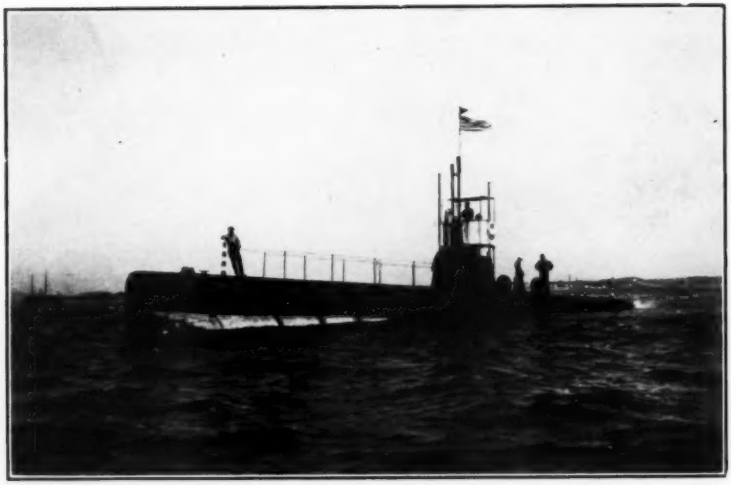


Plate XI.—U. S. Submarine, "Narwhal," Showing the Area of the Spindle Hull which would be Affected by the Depressive Force of Bow Waves.

Fig. No. 2 shows a submarine of cigar-shaped body running in water forty feet deep. The direction of the opposing stream lines is parallel with the true course of the submarine. A is the boat's center of oscillation; B is the bottom or waterbed; C and C′ are areas of maximum streamline pressures, corresponding to the areas of greater pressures in the large pipes to

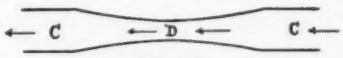


Fig. 1.

which we have already referred; and D and D′ are the areas of reduced pressures—the counterparts of the smaller connecting pipe—and increased speed of flow. When the submarine is running near the surface or with the upper half of her body partly submerged, any impulse tending to change her trim promptly increases the streamline pressure forward at the area C and at the same time brings the stern up, narrows the area for the passage of the water flowing aft and thus diminishes the external pressure or balance of pressure abaft the center of oscillation. Naturally, the boat dives of its own accord, and will continue to do so until the righting arm can

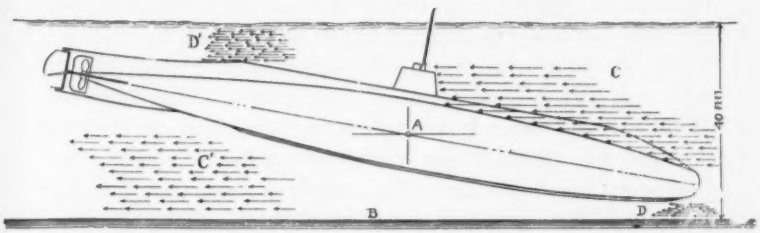


Fig. 2.

exert itself and bring the boat back again to the surface. Again, should the submarine approach the bottom at a downward angle, the area D′ becomes the region of reduced pressure and increased speed of flow on the part of the streamlines, while the area C′ abaft the center of oscillation becomes the region of

maximum pressure; the first tending directly to draw the bow toward the bottom and the second force tending to raise the stern and thus increase the speed and the angular contact with the waterbed. With the ship-shaped hull, none of the body while running on the surface is thus exposed to the initial impulse to dive created by streamline pressures being brought to bear against the upper surfaces as in the case of the spindle-shaped hull and its quickly reduced water-plane area. Of course, if the ship-shaped hull were designed to submerge by considerable changes of trim, the physical conditions illustrated in the case of the submarine would apply equally to the submersible when once under the water. To offset this, however, the submersible is commonly made to dive with only a small change of trim and, when properly constructed, runs under the water upon nearly an even keel. In the Laurenti boats this trim is substantially horizontal, the angular change being nearly zero.

During the official acceptance trials of the *Octopus*, that submarine was required to make at full speed a dive involving a change of depth of thirty feet and to return to the surface in the shortest time possible. The *Octopus* did this in forty seconds, and in making her dive changed her trim to a maximum of 11 degrees from the horizontal. Diagram Fig. 3 illustrates the *Octopus* going through this maneuver—the control of the boat being represented by a course of ideal smoothness. A, A′, A″, A‴ and A⁗ representing this maneuver from start to finish. It will be noticed that, while the *Octopus* changed her depth at her center of oscillation, during this maneuver, but thirty feet, still, a depth of more than fifty feet was required to permit the vessel to swing in. Had the *Octopus* after changing trim to 11 degrees continued straight down for a period of 20 seconds more, she would have reached the hundred-foot depth line. Again, had she continued to change trim up to 20 seconds from the time of starting to dive, at the rate giving her 11 degrees after a run of 10 seconds, the boat would have acquired a declination of 22 degrees as illustrated by C in the diagram, and before she could be checked and

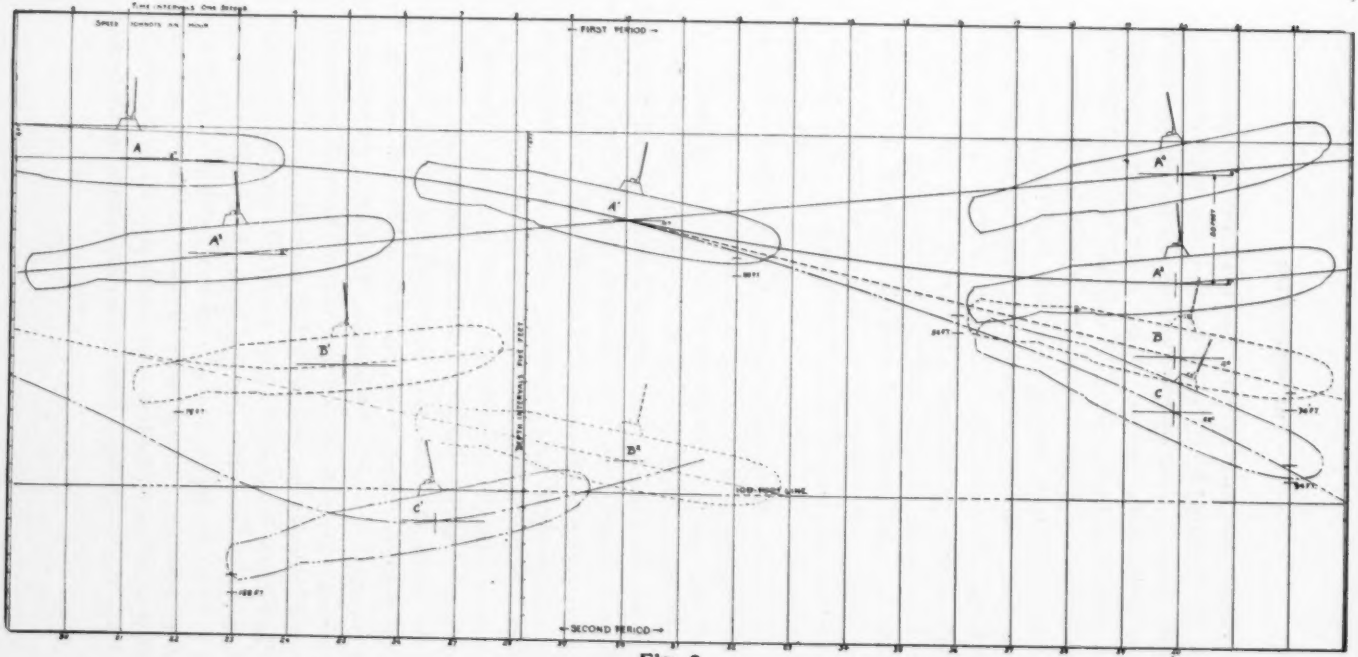


Fig. 3.

started up she would have reached the depth and assumed the position indicated by C'. One can thus see what might easily be the consequences of brief inattention on the part of the man at the diving rudder. None of the recovering angles in the present diagram exceeds 6 degrees, but when the Dutch submarine was undergoing her official trials near Flushing, the diving rudderman, having given the boat too great a down helm, became scared and gave the submarine full rising helm, with the result that she shot to the surface at a recovering angle of quite 30 degrees! The *Octopus* is a boat of 105 feet between perpendiculars, and the accompanying diagram can easily be made to show what would happen if the submarine had a length of 135 to 140 feet. It is not necessary to elaborate upon the advantages of the even-keel boat; they speak for themselves.

During the full-speed submerged runs of the *Octopus* over the measured mile, involving periods from six to eight minutes, the vessel, in the hands of the builder's experts, maintained depth with a variation not exceeding two feet and the trim was held uniform within one degree. The *Hvalen,* after she was turned over to the Swedish Government, and in the hands of her naval crew, was put through a series of full-speed submerged runs extending over a continuous period of more than two hours. During this time the boat was run with frequent changes of direction, these changes including the making of circles and also describing "figure-eights." At no time during this sustained submerged run did the boat's vertical position change more than five inches, and her trim was maintained constantly horizontal. The men at the submerging rudders were of the enlisted force of the Swedish Navy, and their special training for this work had covered but a brief period. The *Hvalen* is a boat of nearly 140 feet in length. The Danish Laurenti submersible, the *Dykkeren,* an all-electrical boat of 130 tons submerged, having a surface speed of 12 knots and a submerged speed of 7.5 knots, has been handled by her Danish crew with similar facility and precision.

Fig. No. 4 is a diagrammatic representation, to scale, of the

cross sections of the *Octopus* and the Laurenti *Foca*. The *Octopus* is built with what is termed a "backbone" (see Fig. No. 5), which extends well forward and well aft above the in-

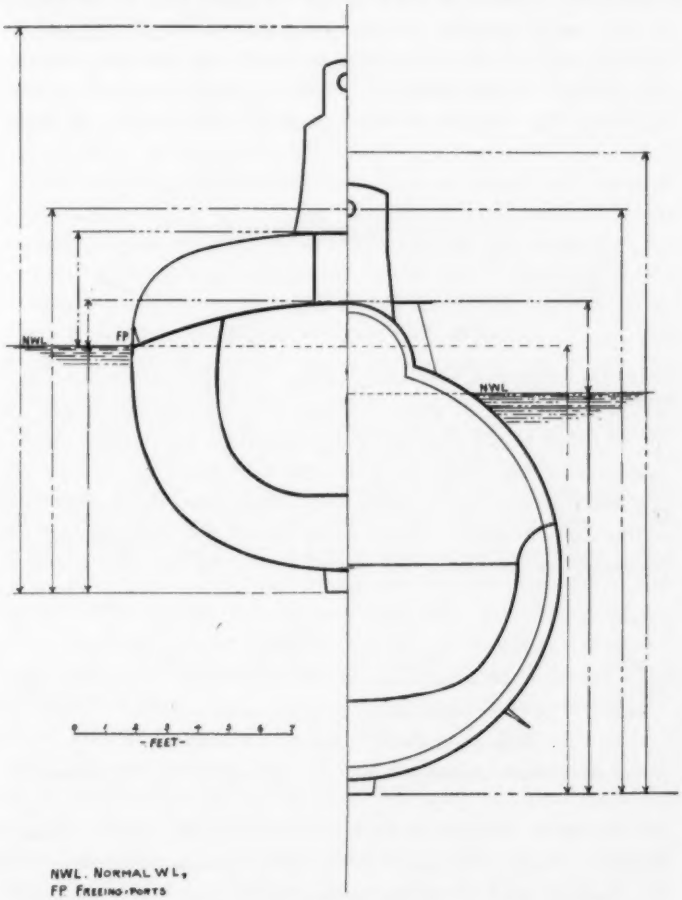


Fig. 4.

terrupted circular sections in order to raise the effective free-board and to increase the buoyancy of the vessel in surface trim. Apart from destroying the integrity and the integral

strength of the circular section, this "backbone" increases the vulnerable area of the main hull open to gun fire. In effect, it is like exposing the boat's spine to easiest attack. The corresponding vulnerable point in the Laurenti boat is the crown of the heavy plating of the pressure-resisting hull, and in Fig. No. 4 this point is made to coincide with the crown of the "backbone" of the *Octopus*. With her maximum of surface buoyancy, the *Octopus* draws 13 feet of water, while the *Foca*,

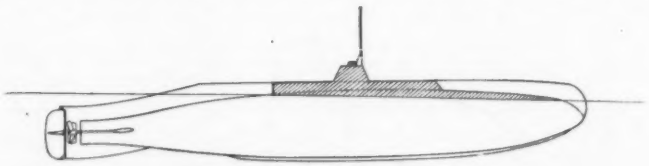


Fig. 5.—"OCTOPUS" IN TRIM.

with her superstructure open, draws in her light surface condition only 8 feet of water. To reduce the vulnerable area of the *Octopus* to the corresponding height of the normal vulnerable area of the *Foca* (Fig. 6) the *Octopus* must reduce her buoyancy greatly and increase her draught something more than a foot and a half. Apart from materially diminishing the buoyancy of the boat, this would make the *Octopus* less safe

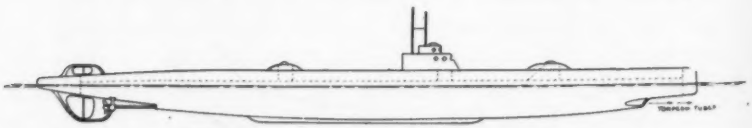


Fig. 6.—"FOCA." SURFACE CONDITION.

Note the safe position of torpedo tubes well below water and safe from immediate damage in case of collision.

and far more difficult to control at high speed on the surface, besides reducing her speed below her normal maximum. We are dealing with a military instrument and not a peacetime craft when we analyze the relative merits of any under-water torpedo boat. The strategic value of light draught and moderate vital target are of the utmost importance, not only because of the wider area of attack and the reduced exposure to gun fire

in the case of the Laurenti boat, but because these vessels must, in turn, seek cover both on the surface and submerged, and the lighter the draught the greater the chance of getting beyond the reach of the modern torpedo-boat destroyer. The *Foca* draws less water than our latest destroyers, and this draught can be reduced somewhat more by letting go the drop keel, if such a measure stand between the submersible and surface pursuit. Again, the Laurenti boat needs less water to completely hide her under the surface.

The normal difference between the unit weight of the ship-shaped section and the circular section has been reduced in the case of the Laurenti boats by the use of hard steel, so that the circular section is not so much lighter as it would be if the ordinary material were used in the Laurenti structure. However, the question of hull weights is susceptible of analysis from two points of view when comparisons are drawn between the spindle hull and the ship-shaped body. As we have already seen, the submarine is emerging from a boat primarily designed to spend most of its time under water and to be strong enough to withstand the impulsive tendencies and the hazards of sudden and dangerously deep submergences. The designers of these boats are gradually trying to bring them more nearly into the class of surface vessels normally. The submersible, on the other hand, has been evolved, in effect, from a surface-going torpedo boat of no powers of defense, and the cover of water has been provided in lieu of armor. The designers of these vessels have worked from the surface downward, aiming to restrict the chances of deep submergence to a minimum and thus to make it less necessary to provide a hull of exceptional strength, the purpose being to give sufficient margin in this particular to meet all reasonable contingencies and to provide an ample time period in which to set in motion the safety features intended to arrest the sinking craft. The diagram, showing the submerging maneuvers of the *Octopus* under varying conditions, points to the necessity of a strength of hull equal to a submergence of at least 200 feet. The suggestion advanced that the ability to sink to the bottom for the

purpose of conserving energy is a desirable feature—hence the pressure-resisting factor of the *Octopus* and kind—is merely an afterthought to justify the use of the circular section. If a vessel wants to save her energy by lying immobile submerged this can be accomplished more economically by anchoring the boat fore and aft at a sufficient depth to give her the desired cover. The other aspect of this question of saving weight for a given strength factor, by using the circular section as against the ship-shaped form, is the more important strategic problem of radius of action and seaworthiness. The boat of circular section requires more power and, consequently, more fuel for the same speed and range of mobility than the ship-shaped submersible. The comparative tables already given plainly signify what this means. The engines for the ship-shaped submersible are lighter, the consumption of fuel less because of the lesser horsepower required for a given result, and this difference in weight tends to offset the heavier ship-shaped hull.

Setting aside these two points of view, there is another one: The ship-shaped hull provides more convenient space inside than a circular section of similar displacement, and, apart from the facility which this affords for the better disposition of machinery and weight, it provides a flexible form which can be far easier molded to meet special local requirements demanding shallow draught, etc. Apart from this, the ship-shaped section of the Laurenti design indicates sooner and over a longer period any tendency toward deformation, and thus gives warning, where none is indicated by the circular section until the instant of collapse. This can be illustrated by a homely comparison. The circular section is under compression and deformation is concentrated at the unsuspected point in the ring where the weakness is hidden. The deformation of the ship-shaped section becomes a matter of tension rather than compression, and the weakness is absorbed for a longer period by the sound elements which constitute the unit section, and this permits of a certain progressive and relative indication of change of form like the strands of a rope under stress.

The relative merits of these two forms must ultimately rest

upon an analysis of the local conditions under which these boats must operate. Fig. No. 7 is a graphic presentation of the twenty-fathom and the fifty-fathom depth lines in operative arcs from the several points on the Atlantic coast of most strategic importance. With the exception of the New England coast, the twenty-fathom line covers an ample operative area for submarine vessels acting purely upon the defensive. On the New England coast the effective area within the twenty-fathom line is not extensive, but the fifty-fathom line encroaches so near to the twenty-fathom line at important points that this lesser depth must not be considered at all so far as safety is assured to a submarine sinking within the likely zone of her military usefulness. The reason given for the Government's requirements that its submarine vessels shall be able to withstand safely a submergence of 200 feet is, that this assures a margin over the depths of the likely areas within which these vessels would be called upon to operate in time of war. To some extent, this is mainly true of our Atlantic coast, but it is not true for the stragetic positions on the Pacific or for the defense of the Panama Canal; nor would this assure safety to the boats called upon to operate in the protection of our naval base in the Hawaiian Islands or for the defense of the prime strategic positions in the Philippines. In most of these positions we are dealing with depths running at short distances into matters of fifty and one hundred fathoms of water.*

In 1909, the British submarine *C11* was run down and sunk off the coast of England in fifteen fathoms of water, and rested on a sandy bottom. Salvage facilities were promptly rushed to the scene of the accident and every effort, that weather per-

* There is some misunderstanding regarding the ultimate strength of the hull of the Laurenti boats. The explosion of gasolene fumes in the superstructure of the *Foca* (1908) gave practical evidence of the amount of shock which the pressure-resisting hull of the Laurenti type will stand. At the moment of that explosion the main hull was subjected to an exterior pressure equal to 12 atmospheres, and yet not a plate or angle of that hull was broken. One plate immediately under the point of maximum violence was a little bit dished, but none of the seams were opened. The *Foca* sank after the explosion simply because the two fire tugs, rushed to the scene, turned their hoses into the boat, pumping water into her until she went to the bottom, believing that there was a fire inside of her which had to be extinguished. This mistake ruined the batteries and occasioned the most costly part of the damage.

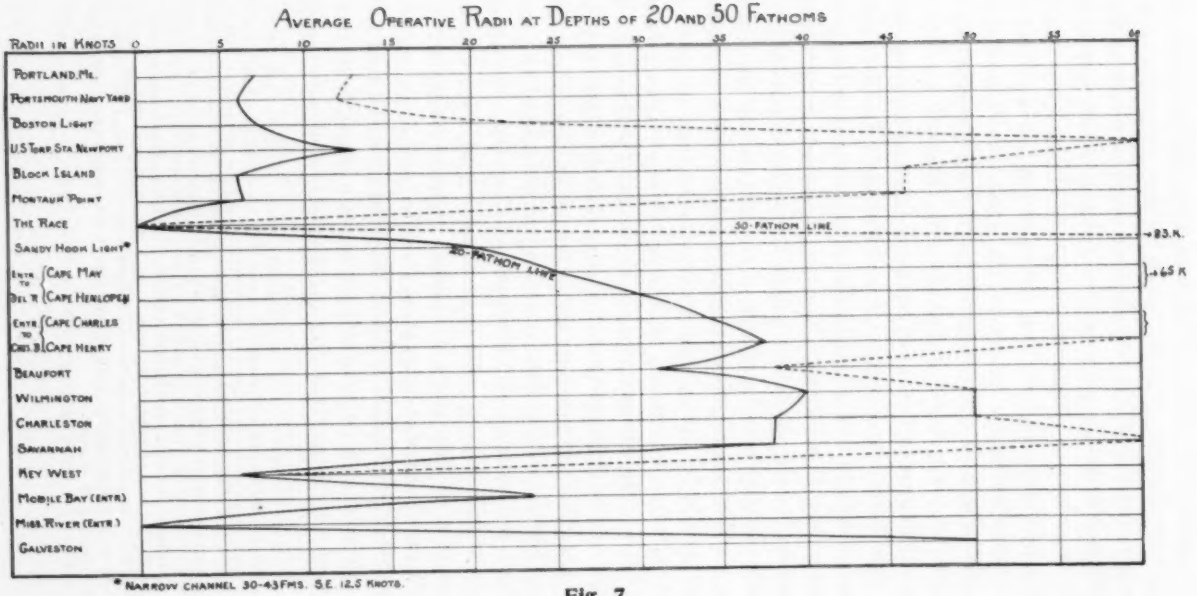

AVERAGE OPERATIVE RADII AT DEPTHS OF 20 AND 50 FATHOMS
RADII IN KNOTS
0
5
10
15
20
25
30
35
40
45
50
55
60
PORTLAND, ME.
PORTSMOUTH NAVY YARD
BOSTON LIGHT
U.S TORP STA NEWPORT
BLOCK ISLAND
MONTAUK POINT
THE RACE
SANDY HOOK LIGHT*
ENTR TO DEL R
CAPE MAY
CAPE HENLOPEN
ENTR TO CHES B
CAPE CHARLES
CAPE HENRY
BEAUFORT
WILMINGTON
CHARLESTON
SAVANNAH
KEY WEST
MOBILE BAY (ENTR)
MISS. RIVER (ENTR)
GALVESTON
50-FATHOM LINE
20-FATHOM LINE
83 K.
65 K
* NARROW CHANNEL 30-43 FMS. S.E. 12.5 KNOTS.

Fig. 7.

mitted, was made to raise the vessel and tow her into port. After efforts extending over nine weeks, the work had to be abandoned, shifting sand having finally buried the boat and rendered further operations impracticable. Looking at the field of activity covered by the diagram, we see that the vessels are called upon to operate in the open sea along the whole exposed line of our Atlantic coast, where from Cape Cod south the bottom is sandy as it was off Cromer, where the *C11* was lost. To any one familiar with the practical difficulties of salvage off our Atlantic coast, the chances of recovering a submarine boat sunk anywhere in twenty fathoms of water are extremely problematical. The boat that can stand a submergence of 200 feet runs as much risk within the twenty-fathom line as the boat designed to submerge to a depth of 150 feet, and the likelihood of recovering either in the open Atlantic at any depth over a hundred feet is not encouraging—in fact, scarcely warrants consideration.

On the other hand, the modern submersible will be called upon to operate further at sea than the twenty-fathom line, and she will have to take her chances with the other types of fighting craft. The best promise of safety lies, when operating upon the surface, in the measure of her reserve buoyancy; and when running under water, her prime factor of safety will lie in the speed and facility with which she can discharge water ballast or release a detachable keel in order to give her instantly a considerable factor of buoyancy. The boat that has the largest reserve of buoyancy will always be the vessel that can best withstand damage leading to leakage, and the longer this buoyancy or any measure of it can be retained the greater the chance of the craft being able to make her way back to port and to safety.

In the comparative table of the *Octopus* and the Laurenti boat, an arbitrary value of 100 was given to the 10-ton drop keel provided for the Laurenti submersible. This is certainly a conservative estimate as against the boat without this feature, because there are likely to be times when the value of this sudden increment of buoyancy can not be overestimated. At all

times, it provides a means for instantly checking the sinking of the craft, thus preventing her from reaching depth dangerous to her hull's power of resistance.

Apropos of this drop-keel feature of the Laurenti boats, some critics have hinted that the weight of this and the permanent keel is necessary in order to give the vessels sufficient stability, the reader being left to infer that nothing of this sort in the way of ballast is needed in boats of the *Octopus* type. The fact is, however, that the *Octopus* has a keel structure and ballast therein weighing together nearly nine tons; but, unfortunately, this weight can not be released to add to the boat's buoyancy in the moment of need. The Laurenti submersibles are designed so that they have a good positive stability under water when all of their ballast tanks are empty and the drop keel released; and it is quite possible to construct these boats without any keel and with sufficient stability for every purpose. In connection with this same question of form of hull and associate stability a good deal of misinformation has been printed. The Laurenti boats, with their superstructures open, are designed to have their centers of gravity under the center of buoyancy when in the surface condition; but the center of gravity can and would be placed over the center of buoyancy with perfect safety should it be necessary to construct a Laurenti submersible of particularly shallow draught. Even if the center of buoyancy were below the center of gravity, and G and B should cross during the period of trimming for submergence, the true basis of judgment at any instant is the value of GM remaining, which is ample in the case of the Italian submersibles. (See Fig. 8.) In the Laurenti boats, as in the Holland boats, all water ballast admitted for submergence lowers the center of gravity, and there is no critical period during the admission of water ballast as is the case with some other boats.

RESERVE OF BUOYANCY.

A good deal of misunderstanding has existed regarding the maximum reserve of buoyancy of the Laurenti submersibles, and some effort has been made to qualify the actual value of

this percentage. If one will look at the half cross section of the *Foca* it will easily be understood that with the superstructure closed, the Laurenti boat has the surface conditions of an ordinary torpedo boat. It is possible, by closing the superstructure, to give these vessels a reserve of buoyancy of from 50 per cent. to 60 per cent. of their surface displacements. This is just as effective as the same measure of buoyancy would be in the case of any other vessel, and would rightly form the basis for judging her sea-going qualifications. With the superstructure open, the water line lies just below the bottoms of the freeing-ports, and the measure of reserve buoyancy—ranging from 20 per cent. to 30 per cent. of the surface displacement—is the maximum reserve of buoyancy of the pressure-resisting hull. This emerged area of the pressure-resisting hull is substantially a counterpart of the protective deck of any other fighting ship, and it would be quite as unreasonable to say that the other fighting ships were not entitled to the reserve of buoyancy represented by the exclusion of water from the protective deck. Even with the freeing ports open, the Laurenti boats have a reserve of buoyancy considerably in excess of the maximum reserve of buoyancy now obtaining in any existing "submarine," and the manner in which still greater buoyancy is assured puts the Italian submersibles in a class apart from all others. In case of collision, the rupture of the superstructure of a Laurenti submersible need not seriously impair the sea-keeping qualities of the boat, the remaining reserve of buoyancy of the pressure-resisting hull being still unimpaired and ample in conjunction with the advantages of the ship-shaped form. Any effort to belittle the advantages of the maximum reserve of buoyancy and the manner in which it is obtained in the Laurenti boats is mistaken energy, because no seafaring man will bother his head about the manner in which this safety factor is calculated so long as the vessel has this measure, which means seaworthiness and the capacity to withstand a good deal of damage before being reduced to the critical stage of less desirable boats of the true submarine class.

SURFACE AND SUBMERGED SPEEDS.

It is an accepted fact that the under-water boat grows in military value as her speeds on the surface and submerged are rationally increased. If we could give to these vessels the same speed submerged that we can now give them on the surface, reserving the qualification that this higher speed under water be of ample endurance and not obtained at a serious sacrifice, then we should have the ideal torpedo craft. It is true that submerged speed enables the under-water boat to cover wider areas for short periods in the final stage of actual attack, and increases the chances of successfully intercepting and torpedoing an enemy under way; but there is a logical limit to this power of submerged attack, and this limit is imposed by the demand for surface speed and the capacity to maintain this speed under all probable operative conditions of the sea.

The boat that can hold the sea and can make a maximum speed of 15 knots an hour is certainly a far more efficient military instrument than another type of under-water boat that can make a maximum of only 11 knots an hour and which has a measure of seaworthiness that can be directly evaluated by the percentage of its reserve of buoyancy and the well-known limitations of the spindle form of hull. Greater submerged speed, in the case of this latter type, is the effort made to offset the surface handicap, and in a craft of this sort it is necessary that she have a higher ratio of under-water speed in order to compensate in part for her surface disabilities. There was a time when the submersible's greater reserve buoyancy interfered with the facility with which the boat could be prepared for submerged work. When this was the case the greater surface speed of the submersible was neutralized in this manner, and the efficiency of the submersible and the submarine in the neighborhood of an enemy was properly measured by the single factor of speed under water. This disability has disappeared to a large extent, and in the case of the Laurenti boats they are quite on a par with the best records of the modern sub-

marine in trimming for submergence; in fact, it may rightly be claimed for these vessels that they handle their water ballast and reduce their reserve of buoyancy with relatively greater facility than any of their rivals, either of the submarine or submersible classes. The *Octopus* has a reserve of buoyancy on the surface of 35 tons, and the *Foca* and *Hvalen* have a pressure-resisting hull reserve of buoyancy of 45 tons. (This does not refer to the still greater reserve of buoyancy of 36 per cent more which the superstructure when closed provides; this additional buoyancy being overcome automatically by the flooding of the superstructure as the lower ballast tanks are filled.) During the official trial of the *Octopus*, starting with the vessel at cruising speed, the time required to pass to the submerged condition—including sealing the boat and trimming for diving—was 4 minutes and 45 seconds. The Laurenti boats, with 28.5 per cent. greater buoyancy to overcome by admitting water, perform the same operation in a little less than 5 minutes. Apparently, the admission of water ballast and the control of the trimming tanks are handled with more ease and speed than in the case of the *Octopus*. By keeping the freeing-ports of the superstructure closed, thus excluding water during the trimming period, it is possible to retain the entire buoyancy of the superstructure while the ballast tanks are filled with the exception of the single main or final submerging tank. By simply opening the freeing-ports this superstructure buoyancy can be speedily overcome without further adjustment of ballast. The advantage of this lies in the fact, that the Laurenti boat in this state is, so far as speed of submergence is concerned, substantially a counterpart of the "awash" condition of the *Octopus*, but with this important difference: the remaining reserve of buoyancy and freeboard enable her to be run safely at high speed under her engines, thus leaving the battery capacity unimpaired and in reserve for strictly submerged work. The original disability or handicap of the submersible as compared with the speed of trimming for submerging of the submarine has thus disappeared, while a state of intermediate advantage has been developed in the case of the Laurenti vessels.

Now, let us see if 15 knots on the surface, combined with excellent sea-keeping qualities, does not make a better offensive boat than the submarine *Octopus* of 11 knots surface speed and nearly three knots greater submerged speed than her Italian rivals. An effort has been made to reduce this problem to a geometrical solution, but Mr. Spear left out of the question the state of the sea, the uncertainties of determining the course and the speed of the enemy, and also the personal equation of

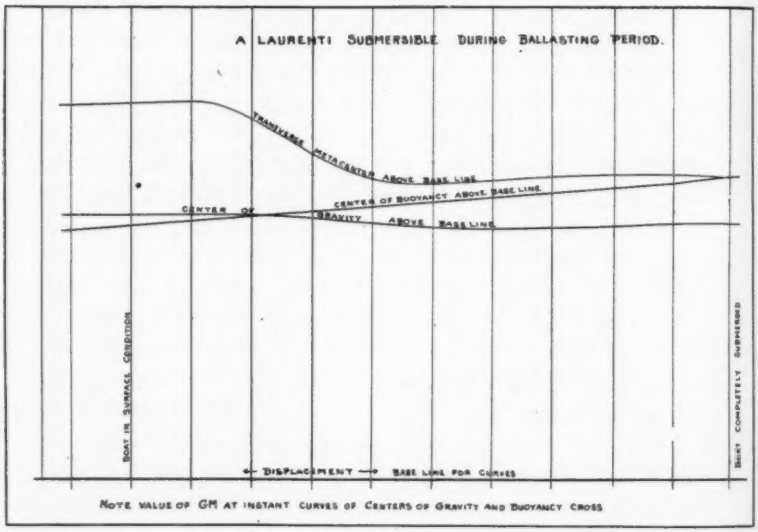

A LAURENTI SUBMERSIBLE DURING BALLASTING PERIOD.
TRANSVERSE METACENTER ABOVE BASE LINE
CENTER OF BUOYANCY ABOVE BASE LINE
CENTER OF GRAVITY ABOVE BASE LINE
BOAT IN SURFACE CONDITION
BOAT COMPLETELY SUBMERGED
DISPLACEMENT — BASE LINE FOR CURVES
NOTE VALUE OF GM AT INSTANT CURVES OF CENTERS OF GRAVITY AND BUOYANCY CROSS

FIG.8

Fig. 8.

the commander of the under-water boat and his capacity to take advantage of the situation as it might develop. The practical problem will always involve several elements of chance, and the author believes that superior surface speed is far more likely to neutralize the difficulties than high submerged speed of necessarily limited endurance. The surface speed will be the means of putting the under-water boat in the best position to make use of her submerged capabilities, and this advantage

becomes more and more pronounced as the radius of action and the field of operations are extended. Submarine vessels will not be expected to operate singly any more than surface torpedo craft. They will work in groups and with satisfactory means of intercommunication both above and below water. They will seek cover under water only when they have got within touch of the enemy—"touch," in this case, meaning the earliest visual knowledge of the advancing foe. Accepting this visual range under average conditions by daylight to to be a distance of five miles, the under-water boat will probably see the masts of her foe before being seen herself, and this would give the submarine vessel time to trim for submergence and get below the surface safely. Suppose the flotilla of submarine torpedo boats has been advised by wireless that the enemy is approaching along a certain approximate course and at a given estimated speed. This information, allowing for a possible error of about ten knots from a twenty-mile radius, would mean that the intercepting torpedo boats might have to cover a distance of from twenty to thirty miles in blocking the on-coming foe. Fig. No. 9 is intended to show the part that surface speed would play in this operation. It is perfectly plain that the submersibles would have a distinct advantage over the submarines. Eliminating the time required to trim from surface to submerged condition—an interval which is substantially identical in the cases of the *Octopus* and the Italian boats under consideration—it is perfectly plain that the submarine is equal to the submersible only when the distance to be covered is considerably shortened, and the vessels are obliged that much sooner to resort to under-water tactics. This aspect of the employment of these boats necessarily implies their use in immediate touch with harbor bases or when operating in restricted waters. This is not the field of usefulness demanded by our geographical conditions, and, therefore, the true submersible is the only boat capable of meeting fittingly the wider requirements that our coast lines impose.

High speed under water is not an unalloyed blessing; it involves some decided disadvantages. The accompanying illus-

trations of the *Octopus* and of the *Dykkeren* while running submerged plainly indicate the magnitude of the tell-tale surface disturbances occasioned by the periscope tubes. This wake, because of its character and possibly its direction, even in a broken sea, would catch the eye of an alert enemy before the submerged torpedo boat had got within likely striking distance. Therefore, to make the best use of submerged speed,

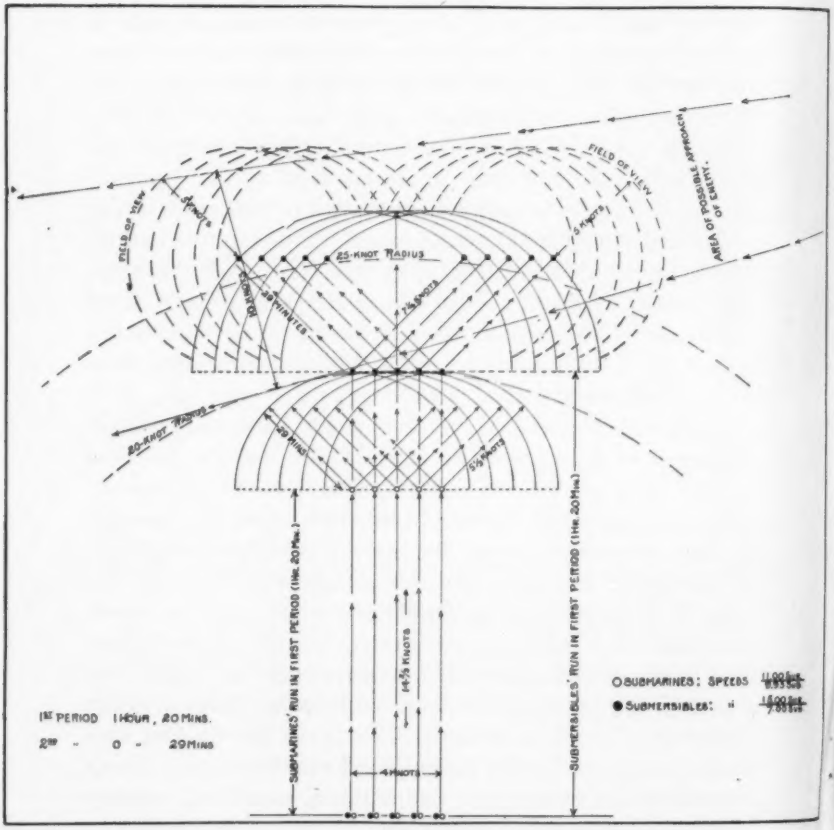


Fig. 9.

The possibilities of submerged speed are not here illustrated. They depend upon the positions which surface speeds allow the boats to take.

Plate XII.—Danish Submersible, "Dykkeren" Submerging. Horizontal Trim About 2½ Degrees by the Head.

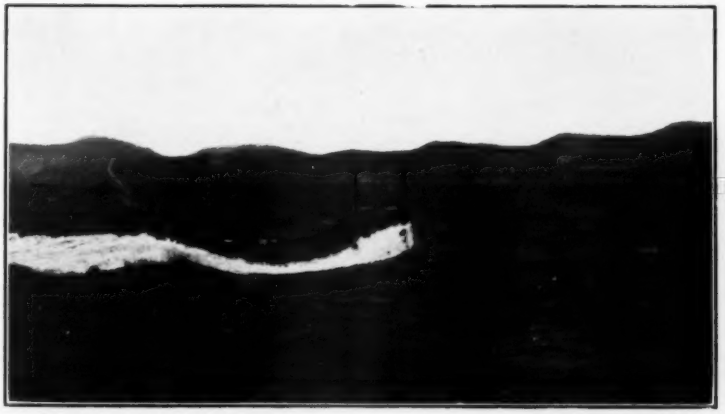


Plate XIII.—Danish Submersible Running at Full Speed Submerged, Trim Perfectly Horizontal.

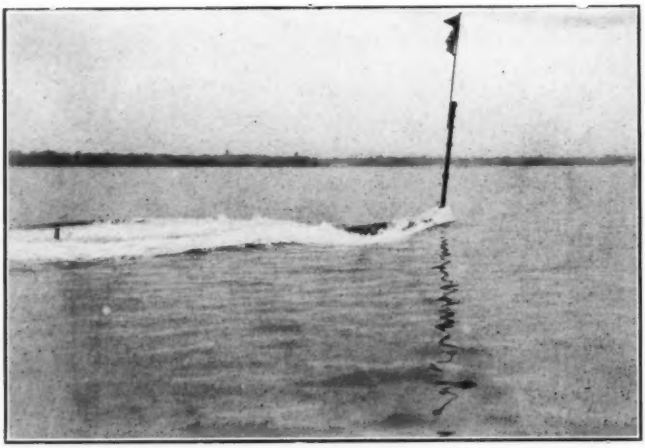


Plate XIV.—Showing Wake of the U. S. Submarine "Octopus," the Boat Submerging at Full Speed.

the submarine vessel must go it blind, after laying her course for her target, from beyond the range of the betraying wake of her periscope. This would be a distance of possibly 2,000 yards or even more, and within this time the enemy might change her course unknown to the submerged boat, thus leaving the latter to spend her energies in the wrong direction at the most critical period of the attack. The under-water boat, per force of these circumstances, must therefore remain to a large extent a weapon of chance; and the nearer she can be carried speedily on the surface to the point of interception, and the better and more covertly she can approach submerged while observing her nearing foe, just so much more likely will be her chances of striking a successful blow.

The submersible of high surface speed is today capable of holding the sea for a considerable period in conjunction with armored squadrons, and there are coöperative possibilities which are very significant, and which will readily suggest themselves to the naval strategist. The submersible can and will act most effectually in "blue water," and that is the logical *local* in which she should be able to intercept the high-speed armored craft of today.

As we well know, here in the United States, official acceptance of any type imposes the realization of the same surface and the same submerged speeds. For a given performance, the submersible calls for somewhat greater submerged displacement, *i. e.*, space for added water ballast to overcome the larger measure of reserve buoyancy on the surface. This factor of reserve buoyancy represents the difference between a tight coat and a loose one which gives the greater freedom of movement, permitting its possessor to exercise his energies with more certainty and force. By reason of this greater reserve buoyancy the submersible is a better sea boat, a safer vessel, and can carry its personnel to the point of action in a far more efficient condition. As we have seen, this added surface buoyancy is easily and quickly offset by the admission of water ballast in the case of the Laurenti boats, making them

comparable in this particular with the submarine of less surface buoyancy.

ADDITIONAL NOTE.

In the November issue, 1908, of the JOURNAL, there was a reprint of Major Laurenti's article published in "Cassier's Magazine" for the same month. This article was originally submitted to "Cassier's" editor in Italian, and the editor erred in translating "stem" into "stern," and explained the mistake on the score that he thought submarine boats, like motor boats, squatted by the stern when running at full speed. It is quite needless to remark that a designer of under-water boats of Major Laurenti's successful experience could hardly have been ignorant of the peculiar tendency of the submarine to plunge by the bow when his submersibles were purposely fashioned and made to operate in a manner to overcome this disability of the rival type.

FUEL-OIL INSTALLATION AND TESTS MADE ON U. S. S. *CHEYENNE.*

By F. F. Ingram, Machinist, U. S. N.

The U. S. S. *Wyoming* is a single-turreted monitor, built at the Union Iron Works, San Francisco, California, in 1901. Length over all, 225 feet 1½ inches; extreme breadth, 49 feet 11¼ inches; normal draught, 12 feet 6 inches; net tonnage, 1,637.27 tons; displacement, 3,128 tons. The armor is continuous fore and aft. The battery consists of two 12-inch breech-loading rifles in an electrically-controlled turret, four 4-inch rapid-fire guns, three 6-pounders, two 6-mm. colts and four 1-pounders.

The ship has two double-acting, vertical, inverted, triple-expansion engines, 17 × 26¼ × 40 × 20-inch stroke, which with the auxiliaries developed 2,451 horespower at 201.3 revolutions per minute with 227 pounds boiler pressure. All the independent pumps, including main feed pumps, are of the improved Dow type. The propellers are 9 feet diameter and 8 feet 4 inches pitch.

Four Babcock & Wilcox boilers are installed, each having thirteen sections. Each section has seven clusters of 2-inch tubes and one of 4-inch tubes, length 8 feet 9 inches, with 36-inch drum. Each boiler has 50 square feet of grate surface and 2,200 square feet of heating surface. Ratio of grate surface to heating surface, 44 to 1; length of grate 6 feet 2 inches.

Two evaporators of 2,500 gallons capacity furnish water to the crew of 230 men and officers; also make-up feed for boilers.

Four dynamos of 50 kilowatts each furnish power for the ventillating blowers, turrets, ammunition hoists, searchlights and for lighting the ship.

During the month of September, 1907, the Navy Department

ordered the *Wyoming* fitted up with an oil-burning plant in order to determine the value of fuel oil over coal on vessels of the Navy. Subsequent to a call for bids the fuel-oil manufacturers of the Pacific Coast submitted various schemes and arrangements for approval. Plans and specifications were furnished those who wished to bid on the installation, each bidder having to comply with the Navy requirements. The space allotted being rather small and existing conditions on a modern man-of-war being much different from those on a vessel of commerce, arrangements for converting back to coal at a short notice had to be considered, thereby causing as slight change in the combustion space as possible. The peculiar condition of the furnaces and ash pans had to be carefully considered. The ash pans are 8 inches below the floor plates in the fireroom, rendering it almost impossible to install a low-pressure system without obstructing the furnace fronts, which would have to be removed in a hurry providing it was found necessary to use coal in an emergency.

Two large settling tanks were installed in the bunker space forward of the fireroom, each tank having a capacity of 243 barrels or 10,306 gallons. A third settling tank was installed on the port side aft of the fireroom space and forward of the engine room, this tank having a capacity of 104 barrels or 4,368 gallons. In addition to these tanks oil is carried in eighteen double-bottom compartments under the engine-room and bunker spaces. No oil is carried in or under the space occupied by the fireroom compartment, these compartments being reserved for boiler reserve-feed water. A coffer dam 2 feet wide intervenes between the oil and water bottoms, making it impossible for oil to leak through into water compartments without being detected. The coffer dams are fitted with man-hole plates to allow for inspection. Ventilating pipes are also fitted to the coffer-dam space and carried up to superstructure to allow for escaping gas to reach the atmosphere. A 4-inch ventilating pipe is connected to all the settling tanks and double bottoms, carried up to the superstructure deck, and fitted with a return bend so as to prevent water reaching

the oil tanks. Sounding pipes are fitted to each oil bottom and settling tank, with plates on main deck for sounding.

A 3-inch filling line extends on either side of the ship under the main deck frames from forward to after oil tanks and bottoms, with a cross-over connection to allow oil to be taken at either port or starboard side of ship. Cut-out valves are fitted to each settling tank and double bottom so as to prevent overflow in filling the tanks. A 3-inch independent suction line is also installed connecting all the oil bottoms and settling tanks to the fuel-oil pumps. Oil can be pumped from any of the oil bottoms through the suction line and discharged into the filling line to the settling tanks without interference with the fuel pumps used on the burners.

The quantity of water found in fuel oil in California varies from 1 to 5 per cent., therefore requiring the oil to be transferred from the double bottoms to the settling tanks and allowed to settle for 36 hours before using. The specific gravity of water being greater than oil causes the water to collect at the bottom of the tank, where means are provided to pump out the water. A hand pump is fitted on deck with a suction to the lowest part of each settling tank. The pump discharges into a funnel, where the man pumping can see when all water is removed and oil appears in the funnel.

In December, 1907, after a careful investigation of all the oil-burning devices submitted to the Navy Department, the Staples & Pfeiffer Company, of 102 Stuart Street, San Francisco, California, was awarded the contract to furnish their oil-burning plant, the same to be installed under the supervision of Commander C. A. Carr, U. S. N., Inspector of Machinery at the Mare Island Navy Yard.

A 12 × 12 × 16 duplex, double-acting Clayton air compressor, capacity 830 cubic feet of free air per minute, a cylindrical tank 9 feet long by 40 inches diameter was fitted in the compressor room as a reservoir for the storage of air A 6-inch pipe was run from the storage tank to the fireroom for the burners. A steam connection was also made from the auxiliary steam line to the 6-inch air line so as to be able to

use either steam or air for atomizing purposes. Two oil pumps, $4\frac{1}{2} \times 3\frac{3}{4} \times 4$ inches, having a capacity of 56 gallons per minute, the usual amount of complex attachments, consisting of automatic regulators, heaters, strainers, by-passes and safety appliances usually found in a first class oil-fuel plant; a third or ballast pump is also fitted as a distributing medium from the double bottoms to the settling tanks. The suction of ballast pump is connected to the main suction line and discharges into the filling line connecting with all the oil bottoms and settling tanks.

A great deal of difficulty was experienced in getting the piping in place owing to the fact that pipes passing through compartments and bulkheads must be made watertight. The height of the protective deck interfered to a certain extent, causing the boilers to set so low that the ash pans are 8 inches below the fireroom floor plates, the furnace doors being so low that the attendant must necessarily get down on his hands and knees to look into the furnaces. The bearing bars and grate bars were removed, leaving one common combustion space. The false ash pans were also removed, and a layer of magnesia, 4 inches thick, was placed over the inner bottom in the combustion space; then a layer of fire clay spread over the top of the magnesia, about 1 inch thick; a layer of fire brick was then placed on top and embedded in the fire clay, thereby completing the bottom of the furnace. Bridge walls were constructed at the back for the flame to strike against. Walls were built in front of the ash-pit doors with pigeon holes to retard the incoming air and also superheat same before mingling with the gases. Slides were fitted in the ash-pan doors to allow of opening and closing as amount of draft required. Two burners were placed in each boiler and in such a manner as not to interfere with the opening and closing of the furnace doors. The atomizing line consists of a 2-inch galvanized pipe leading from the main air line to the inside of the combustion space and carried across the furnace front in the inside, thereby causing the atomizing agent to become superheated before coming in contact with the burners. The atomizing

line projects through the furnace front in the center and branches are then taken to the burners. The burners are inserted between the furnace doors and allow the doors to be opened at will for examination of interior of combustion space. The oil piping is carried from the oil pumps in a 1½-inch line to the fireroom, where a strainer is fitted, before entering the superheaters. A by-pass is also fitted in connection with the strainer to allow for changing same without shutting off the oil supply to the burners. A 4-inch boiler tube is used as a superheater. There are caps on each end of the boiler tube, with stuffing boxes in each end. The 1½-inch oil line passes through this 4-inch tube and steam is admitted to the intervening space between oil pipe and 4-inch tube. A pressure of 50 pounds steam is maintained on the heater at all times.

On October 6, 1908, the *Wyoming* was placed in full commission, with Commander J. J. Knapp, U. S. Navy, in command. A crew was sent from the recruiting ship at Mare Island. Neither officers nor men had any experience with fuel oil. The only person on board having any such experience was Warrant Machinist Fred F. Ingram, U. S. Navy, who installed the plant under the supervision of Commander C. A. Carr, U. S. Navy. Very little difficulty was experienced in getting fires lighted. Some few changes were suggested and proved very beneficial. Owing to the fact that the men were not accustomed to burning oil, a certain amount of care had to be exercised in regard to safety. Smoking was prohibited below, and after a few watches everything went along all right. Scale in the piping at first caused a great deal of trouble, clogging both air and oil openings to the burners. At first a great deal of smoke was emitted from the stack, but after the men began to get sufficiently acquainted with the burners to regulate them smoke was rarely seen except when changing burners. A great deal of water was encountered in the first oil furnished, it being of a poor quality and very dirty.

The quality of oil used during a trip from Mare Island to San Francisco was 18.5 specific gravity on the Baume scale, free from water and sediment. Good results were obtained.

After trying various pressures on the oil and atomizing line, it was found that best results were obtained at a pressure of 50 pounds on the oil and 70 pounds on the atomizing line. These pressures, with the oil at a temperature of 230 degrees and air at a temperature of 900 degrees, gave a soft, white incandescent flame, with the entire furnace at a glowing white heat. Steam could be easily maintained at a steady pressure without forcing the burners. Upon coming to anchor the furnaces and tubes were very carefully examined. No deposit of carbon or asphaltum was in evidence. Slight changes had to be made in the arrangement of the bricking of the furnaces, as the return flame caused an undue heating of the furnace fronts and raised the temperature of the fireroom. A series of brick walls extending the height of the burners overcame this difficulty and also superheated the incoming draft. Salt and broken bottles were used freely in the brick work, which, when heated, becomes one mass of glazed walls. Baffling was also found necessary to prevent the flow of unconsumed gases burning in the uptakes. After baffling was put in place the temperature of the combustion chamber increased and the heating in the base of the stack was not noticeable. It was found that with forced draft the gases were driven with such velocity that combustion did not take place until the gases reached the stack, causing the paint to blister and heat the stack to such extent that kinks were noticeable owing to the tightness of the guys or stays.

The amount of air admitted to the furnaces is a very necessary adjunct to a furnace from a point of economy, as too much air admitted causes a roaring noise as the velocity of incoming air drives the gases unconsumed to the uptakes. Smoke is then emitted, which is always a sure sign of imperfect combustion.

On December 31, 1908, the name of the U. S. S. *Wyoming* was changed to U. S. S. *Cheyenne,* by order of the Navy Department, and orders were received to proceed south, to conduct a series of tests previously ordered by the Bureau of Navigation. A great many burners were tested at various times. Both inside and outside mixers were used with varying results.

There are a great many devices on the market for the burning of fuel oil. The oil found in California and many western States contains asphaltum base, which makes it impossible to use mechanical burners, owing to the fact that asphaltum and carbon form readily when heated up to a high temperature and close the small appertures in the mechanical burners, necessitating cleaning and removing very frequently, causing a fluctuation of steam and a great deal of annoyance to the attendant. With Pennsylvania and Ohio oils this difficulty is not experienced, owing to the fact that paraffin is the base, and superheating the oil assists its combustion. The type of burner used on the Pacific Coast is adapted to both paraffin and asphaltum bases.

There are two general types of burners used on the Pacific Coast: what is known as an outside mixer, or atomizer; also an inside mixer, or vaporizer. Both have their relative merits and deserve mention.

First. The outside mixer or atomizer is very simple in construction. There are fewer parts to get out of order, free from internal deposit of carbon and asphaltum in the oil passage, the oil being kept under pressure until it reaches the combustion chamber, when it is released at right angles to the atomizing agent in the open combustion chamber. There are no parts to get out of order or work loose, consequently a higher rate of efficiency is obtained. No adjusting of parts is necessary except to vary the amount of oil to suit the load being carried. With outside mixers the oil and atomizing agent do not come in contact with each other until they reach the combustion chamber, and a small percentage of water in the oil does not interfere materially, as the water contains enough oil to keep the burner lighted; the oil being superheated to 230 degrees in the burner immediately evaporates any water that is held in suspension in the fuel oil. With outside mixers there is no sputtering or series of explosions, so objectionable and annoying to the attendant. Fig. 1 shows an outside mixer or atomizer of the type used in the U. S. S. *Cheyenne.* This burner has given very good results. The oil is kept under a

constant pressure of 50 pounds and the atomizing agent at 70 pounds. The oil passes through a heater that has a steady pressure in its jacket of 50 pounds per square inch. The temperature of saturated steam at 50 pounds pressure equals 281 degrees F., consequently the oil is kept at a high temperature. It is further heated by passing through the superheater 3, in the burner, where the atomizing agent surrounds the oil. This further adds to the already superheated condition of the oil, and upon being released in the combustion chamber at right

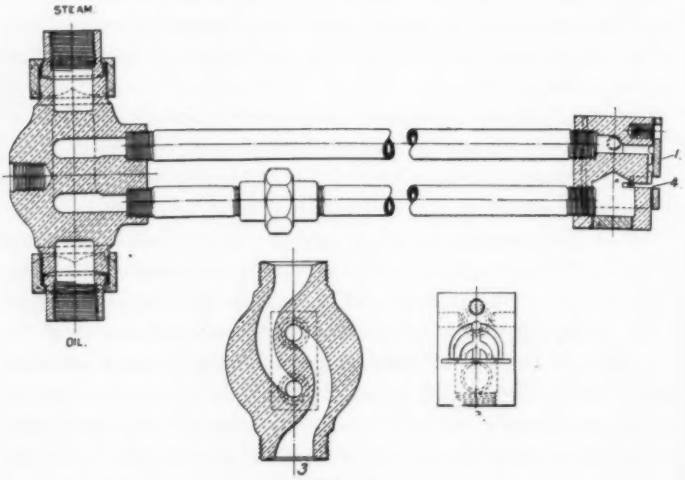


Fig. 1.
INGRAM BURNER.

angles to the atomizing agent it instantly becomes a gas and burns with a soft, white incandescent flame, almost noiseless except when being forced. 2, shows the head of burner. When the oil is delivered through the small hole 5/32-inch in diameter, the oil is retarded by the baffle plate 1, and forced through the circular passages into the annular opening 4, where the atomization agent comes in contact with the oil; a handle is also fitted to these burners to facilitate handling when hot After being placed in position the handle is detached. To

clean these burners, remove the baffle plate and clean the obstruction from the circular grooves; with good oil these burners do not require to be changed for weeks.

The U. S. Tug *Navajo* is fitted with the Ingram type of outside mixer or atomizer, and on a recent trip from San Francisco to San Diego, a distance of 500 miles, these burners were not removed from the furnaces nor did they require cleaning during the entire trip. Steam was maintained at a steady pressure and smoke was not seen during the trip, showing that combustion was perfect. The U. S. Tug *Unadilla* and U. S. Monitor *Cheyenne,* also steam launch and galleys, are equipped with Ingram burners. Coal was not used on the U. S. S. *Cheyenne* for four months previous to going out of commission at Mare Island.

Second. With inside mixers or vaporizers the steam or other atomizer comes in contact with the oil in the internal portion of the burner. The oil is contained in a pipe $\frac{3}{8}$-inch diameter inside of a pipe 1 inch in diameter. The steam surrounds the oil pipe, therefore heating it to a very high temperature. The oil passes out at the small holes in the mixer where it comes in contact with the steam. It passes through to the mixing space where it is given a circular motion by rifling in the mixer head. Then it passes to the burner tip and is admitted into the combustion chamber in the saturated condition. A soft, white incandescent flame is the result. With careful regulation the inside mixer gives excellent results, but when forced it has a tendency to smoke and show a red flame, because more oil is being admitted to the burner than it can vaporize, consequently a great deal is wasted. There are a great many parts to the inside mixer. It requires a certain amount of skill to take apart and put together again. An objectionable feature is the packing contained in the burner, as the high temperature of the metal has a tendency to burn the packing thereby causing it to leak steam into the oil passage. A great deal of care must be exercised with inside mixers, as any increase or fluctuation in the pressure changes the mixture of the gas and causes it to go out or to sputter or give off a series of explosions, giving notice that it needs attention.

With inside mixers water in the oil has caused a great deal of annoyance, and it has been proven that oil containing 1 per cent. of water cannot be burned with inside mixers without giving a great deal of trouble and smoke, sputtering, and finally going out. In another boiler using the same oil, same atomizing agent, same conditions except that an outside mixer was installed, no change seemed to occur except that the oil valve had to be opened a little more.

An inside mixer type of burner of the Staples & Pfeiffer design has been used extensively on the Pacific Coast for a number of years and given excellent results. Troubles encountered with this type of burner are not always due to the burner, but in many cases due to faults in other parts of the installation. Great care should be taken to see that an excessive amount of air does not enter the combustion space, causing an undue amount of waste by heating the surplus air. Too little air will be easily detected by the furnace showing a dull red flame, indicating that not enough oxygen is supplied to consume the gases.

With fuel oil a steady pressure can be maintained and hence more uniform and efficient results obtained at the engines than with coal. With coal it is necessary to open the furnace doors at short intervals to allow for coaling, slicing and raking; also cleaning fires, causing a fluctuation in the steam pressure, besides allowing a great rush of cold air to strike the heated tubes, causing unequal strains on parts of the boiler exposed to the cold air, thus causing warping and bending of tubes, and leaky tubes. With oil fuel it is not necessary to open the furnace doors, as peep holes are provided in the furnaces over each burner, enabling the attendant to watch the burner. It is very desirable in ships of war to get up steam in a hurry. On one occasion a boiler was connected up in 26 minutes from cold water. There is also the advantage that on cutting a boiler out or securing boilers, when fires are extinguished the fuel expense ceases and the boiler ceases to steam. Valves can be closed and ash-pan secured, damper closed, there is no further trouble; no fires to haul, no ashes to care for and get in bilges

to stop up strainers; no grate bars to renew; no tubes to sweep, nor paint work to scrape on coming to anchor. With oil all such worry and labor is avoided and the labor for coaling ship is reduced to two men. With oil a hose is led to the filling valve on the outside of the ship, the pump is started and a manipulation of the valves is all that is necessary. Three hours is all that is needed to supply 1,485 barrels of oil, where in a former commission it required the ship's company all day to take on 388 tons of coal, besides a whole day to scrub and clean ship, and the next day for the men to remove the coal dust from their bodies and clothes.

INSTALLATION.

There are no given or set rules for the bricking or arrangement of the interior of the furnaces. With tubular boilers the custom is to have the grate bars in and construct a blind wall under the front row of grate bars, near their further end to prevent the air passing into the back connection. The grate bars are covered with a layer of fire brick. A few bricks are omitted directly under the burner to allow of sufficient air to pass in to form combustion. It also tends to superheat the incoming air. With water-tube boilers the usual practice is to remove the grate and bearing bars and construct a bridge wall in front, with occasional holes to allow sufficient air to pass to the combustion chamber. In large combustion chambers pyramids of open brick work are often constructed, as they assist combustion by giving off a glowing heat. No general rule can be followed for installation, as conditions vary. Chimney heights have to be taken into consideration. Baffling and bricking should be done to suit the individual plant.

A great deal of fuel is needlessly expended in smoke and on account of defective installation. When the amount of air supplied is not sufficient a certain amount of gases must pass off unconsumed into the atmosphere in the form of smoke. If we could remove the carbon and sulphur from the smoke we would have smokeless burners. Usually it is the fault of the

installation, and not the burners, that causes smoke. If the burners are turned down sufficiently, so as to eliminate smoke, you may know that the supply of air is nearly right, but this will produce a drop in the steam pressure, as the oil must be turned on smoke or no smoke. Many plants are installed without regard to economy because a contract has been awarded to the lowest bidder, and he in turn installs the plant in the most convenient manner to suit his own pocket-book, regardless of economic results. A certain evaporation efficiency is usually guaranteed per pound of oil, but few plant owners care to go to such trouble or expense to prove the truth of this boasted guarantee. A reducing valve should be installed on the steam line leading to the burners, thereby causing steady and even pressure at the burner. With the boiler pressure on the burner a slight fluctuation in the pressure causes a change in the adjustment. After months of experimenting at different pressures on oil and steam, a constant pressure of 70 pounds on the steam and 50 pounds on the oil gave good results, and these pressures were adopted by the inventor. After the burner is once adjusted to the load no further adjustment is necessary. All fittings on pipes 3 feet and up should be made up with screwed flanges beaded over into a counterbore. Gaskets of rubber should not be used, as the oil dissolves the rubber and causes leaky joints. A piece of No. 1 canvas shellaced on both sides and made up wet will never give any future trouble. Seven threads on all piping where oil passes through should be made up with shellac. Means for getting rid of an accumulation of water in the oil should be provided, as a great deal of trouble is caused by water; sputtering and intermittent explosions are the results.

Foreign substances in the oil cause a lot of annoyance to the attendant. Strainers or filters composed of fine gauze wire around the basket of the strainer, and placed so as to be easily gotten at to clean these strainers, should be on the discharge line from the fuel-oil pump to the burner. Magnesia and dirt are hard to burn and are prime factors in plugging up the holes in the burners.

A separate line should be employed for filling tanks and double bottoms on ships, thereby not interfering with the suction from any tank. The suction line should be placed as low down as possible, and the pumps should be installed in such a manner as to prevent any great lift on the suction side. Means should also be provided for proper ventilation to all tanks and vents should be carried sufficiently high so as to avoid ignition of vapors. Gate valves on all suction lines prevent undue friction, as they can be opened their full area. Globe valves are not so well adapted to oil as gate valves, as cold oil flows very sluggishly. Restricted areas and short bends in pipe and fittings cause undue friction, thereby retarding the flow of oil. Ships that cruise in Northern latitudes are fitted with a steam coil in their fuel tank, so as to assist the pump in delivering the oil to the burners by rendering it more fluid. Great care should be exercised in regard to leaky joints, as disastrous results might occur from an accumulation of gas in bilges and bunkers. The oil lines in the fireroom space should be examined once each week for leaks. Great care should be taken around the furnaces while changing burners to see that no oil valves are left open. Burners should never be allowed to remain in dead or cold boilers, as the oil valve might accidentally be opened, causing the oil to flow into the furnace unnoticed; a formation of gas and an explosion might be the result, due to someone's carelessness. The usual method is to have a rack. When the burners are removed from the boilers they are cleaned and placed in the rack, where they remain until required for use.

Combustion of petroleum, like that of all gaseous fuels, can be carried on with a much higher degree of efficiency than that of solid fuel. While the calorific value of 1 pound of oil is as 1.37 to that of a pound of bituminous coal, and 1.63 to that of a pound of anthracite coal, in actual practice it is found that 1 pound of oil is fully equivalent to 1.8 pounds of coal. There are three forms of advantage in the use of oil apart from the direct cost of fuel. These relate to the economy of labor in attendance at the boiler, owing to the larger consum-

ing capacity of a boiler, because the heat is applied continuously without any interference or interruption compared to that of applying the coal on a furnace of an ordinary boiler. There is the absence of smoke and cinders, or waste of any kind, compared with the smoke and ash and refuse left from coal, and lastly, its cleanliness together with its ease of manipulation at the furnace.

In oil-fuel burning a high furnace temperature gives more perfect combustion, and it requires less force to atomize the oil. With a properly arranged furnace, very little steam or air pressure is necessary for atomization, because the heat from the glowing brick work immediately converts the petroleum into an oil gas, and sufficient oxygen being admitted to the furnace is all that is necessary for complete combustion. The ash pan should be arranged with a door that could be adjusted to suit the amount of air necessary for combustion, which can be easily seen by an experienced fireman. A clear, white, incandescent flame burning up to the tip of the burner denotes proper combustion. Means should be provided to heat the incoming air before coming in contact with the oil gas. Different methods are used to obtain these results, but general practice will determine the most suitable arrangement for each individual installation, as conditions differ in most plants that were originally equipped for coal.

Smoke consists of the combustible and incombustible products evolved in the combustion of fuel oil, and is composed of such portions of hydrogen and carbon of the fuel gas as have not been supplied or combined with oxygen, and, consequently, have not been converted into a carbonic acid; the hydrogen so passing away is invisible, but the carbon, upon being separated from the hydrogen, loses its gaseous character and returns to its elementary or organic condition, that of a black, pulverized carbon, and as such becomes visible and passes up the stack in the form of unconsumed carbon, or smoke. Carbonic acid is the result of perfect combustion; carbonic oxide is that of imperfect combustion. One pound of carbon combines with 2.5 pounds of oxygen and produced 3.5 pounds of carbonic acid.

An insufficient supply of air causes imperfect combustion and causes the unconsumed gases to be given off in the form of smoke, while an excess of air causes a waste of heat.

The volatile products of oil gas in the furnaces are "carbureted hydrogen" and olefant gas, which, upon combining with the oxygen of the air, becomes carbonic acid.

Theoretical or complete combustion teaches us that for every atom of carbon and for every atom of hydrogen there should be at least one atom of oxygen brought into contact with each other, and then subjected to a temperature sufficiently high to ignite them. The only way carbon can be burned is by diffusion of the gases, whereby the molecules are brought into close contact with each other. But this diffusion requires time, as the gases are constantly being cooled by the incoming air, and carried away to the uptakes before being sufficiently heated; and, not reaching the point of ignition, the particles pass up the stack in the form of dense black smoke. An excess of air entering the furnace is a waste, especially with oil fuel, as too much air cools the furnace sufficiently low to retard the proper commingling of the gases. The incoming air should, if possible, be superheated before coming in contact with the oil gas. A portion of the air is heated by coming in contact with the brick work inside the furnace, which, under ordinary conditions, is very hot. Therefore it is evident that with the fuel-oil installation forced or induced draft is not to be considered economical, as the open furnace allows the incoming air to flow to the combustion chamber with such velocity that the gases are cooled below the ignition point and pass off unconsumed. Only the gases that are deflected and baffled by a series of brick work in the combustion space are ignited; consequently a large volume of smoke is emitted. A short, white incandescent flame indicates good combustion, as the cooled gases from perfect combustion are colorless.

While it is possible for the diffusion of gases to be completed before combustion begins, as in the Bunsen gas burner, the difficulties naturally disappear, and there is readily attained a very short flame, which, moreover, is incapable of depositing

soot even on a cold object. In the case of liquid fuel, which is incapable of vaporization, the diffusion and ignition must occur simultaneously. With such a fuel there is bound to be considerable flaming, and a long tongue of flame issuing from a burner with great velocity will, sooner or later, prove disastrous to the back sheet of the combustion chamber. Where stay-bolt nuts are exposed to the flame they should be protected by porcupines held in place by set screws, or, what is still more desirable, a lining of fire brick. The point of the flame is always the hottest, and if it impinges on a tube or back sheet it will burn it. The flame being so intense, and the circulation in that portion of the boiler very slow, a film of steam forms between the tube or sheet and prevents radiation of the metal, thereby causing overheating and bulging of the back sheet of the combustion chamber, and, in many cases, in water-tube boilers the tubes become warped and bent due to the flame impinging on the tube. Several naval vessels have had tubes burned out on their trial trips due to this cause. These ships were fitted up with oil as auxiliary to coal, similar to what has recently been the practice in the British Navy. The process of burning oil in this way in combination with coal can not be considered satisfactory as far as combustion is concerned. The oil is sprayed on top of the bed of coal. The opening and closing of furnace doors allow a large quantity of air to enter the furnace, cooling down the gases already in the process of diffusion below their ignition point. Also the gases being distilled from the coal suffer from the volumes of excess oxygen being admitted to the combustion space. Great danger of back flaming while working the fires causes the firemen to neglect the fires to a great extent and allows holes to be burned in the back of the furnace, allowing free air to enter without passing through the fuel bed, thereby doing no useful work, but causing a waste of fuel and energy.

SETTLING TANKS.

Where oil is carried in double bottoms, and required to be pumped into settling tanks, some provision should be made

for the getting rid of the water precipitated. A convenient and safe method is to have a hand pump connected up on the main deck from the bottom of the settling tank, and discharge into a funnel and carried overboard. Then when water accumulates in the settling tank it can be pumped independent of any pump that might be on the suction line to the burners. When the water is nearly gone oil will appear in the funnel, showing that the water has been displaced, and is a sign to stop pumping. In every oil-fuel installation special provision should be made for the removal of the water that will collect from various sources at the bottom of the settling tank. Even a small amount of water pumped to the burners will interfere with the efficient and satisfactory work of an oil-fuel installation. As it is essential with every boiler plant to secure a uniform if not a large output, the annoyance and evil of occasionally pumping water rather than oil to the burners can not be overestimated. The trained and intelligent fireman can always tell when water is present in the oil; a slight sputtering and series of short explosions gives warning; if attention is not given the oil system the fire will keep going out. The trained fireman proceeds to drain his heater and get rid of the accumulation of water precipitated caused by the heating of the oil. Therefore it is apparent that more skill and intelligence is required to burn fuel oil than coal. Cheap and careless help around an oil-burning plant is liable to prove very expensive in the end.

PETROLEUM.

Petroleum is the name given to crude oil found in the earth at various depths from 600 feet to 4,000 feet. Great oil-producing fields are located in Ohio, Texas, Pennsylvania and California, Russia, Australia, Canada and Borneo.

Petroleum exists in three elements: 1st, in the gaseous state or natural gas; 2d, as a liquid, as it flows from the wells; 3d, in the solid state, as bitumen. In the pitch lakes of Nicaragua is found the petroleum in the solid or evaporated form, caused by the action of the sun dispelling all the volatile substances

and leaving the residue or bitumen to become solidified. This bitumen or asphalt is used to a great extent in what is known as petrolithic pavements for street use. It is very resilient, and almost noiseless during the summer months. As winter approaches it becomes very hard and slippery when wet and accompanied by frost. It is the universal pavement for semi-tropical regions. In the gaseous state petroleum or natural gas is found in Pennsylvania and Ohio, where it is used for fuel purposes for manufacturing establishments. Many towns in Ohio and Pennsylvania depend entirely upon this form of gas for all purposes—lighting, cooking and heating. This natural gas is given off the petroleum in the bowels of the earth, where it must undergo a very high temperature to generate the natural gas of varying pressures. It is not unusual to find gas wells producing gas at pressures varying from 50 to 150 pounds per square inch. The gas is piped through large mains to the nearby towns and cities, and delivered to the consumer at a very low cost. An instance has been recorded in Pennsylvania where natural gas was delivered at the furnace at 10 cents per thousand feet.

Petroleum in its liquid form, as it flows from the wells in its natural state, contains various chemical combinations and compounds. Petroleum varies greatly in its percentage of carbon. The usual percentage of carbon in California oil is about 85 per cent. carbon and 15 per cent. hydrogen. The specific gravity varies with the typography of the country where found. In California and Texas oils the residue or base is asphaltum or bitumen, a thick, semi-fluid, tarry substance, becoming hard after exposure to the atmosphere. Oils found in Pennsylvania have a paraffin base or residue. Upon exposure it becomes crystalized but with the application of heat it dissolves readily. It is used in the manufacture of candles and wax tapers. Crude oil is understood to be the natural oil, neither strained nor treated, free from water and sediment and of a specific gravity of 43 to 46 degrees Beaumé. Naphtha is required to be water-white, and specific gravity of 68 to 72 degrees Beaumé. Kerosene to have a flash point not below 110 degrees F.; while

residue is understood to be the refuse from the distillation of crude petroleum, free from coke and water and sulphur, and specific gravity of 16 to 20 degrees Beaumé.

Crude oil is recognized by color, smell, specific gravity, flash point and firing point or heat test.

Color.—The color of California crude petroleum ranges from a deep black to water-white, the usual color being black or dark brown. This dark color is due to the asphalt, and changes to green of more or less brilliancy on removal of this substance. A few of the California wells contain little or no asphalt and are greenish in their crude state, when even some of the lighter asphaltic oils show a green tinge. The color of the crude oils is uniformly green, but not very pronounced. The bloom on refined oils follow the usual rule; white in the naphthas, blue in the kerosenes and lighter lubricating oils, changing to green in the heavier lubricants and vaseline.

Odor.—The odor of the heavier oils is highly characteristic, mild and rather sweet; very rarely sulphurous. The foul odor of the Texas, Canada and Ohio oils is entirely absent. Many of the lighter oils are sweet, resembling Pennsylvania crude in this respect. In brief, the odor is very seldom objectionable.

Specific Gravity.—The specific gravity ranges from 3 to 58 degrees Beaumé scale, but most of the producing wells in California fall between 13 and 35 degrees Beaumé. The production of the entire State would average about 16.5 degrees. It will be noted that the specific gravity is very low compared with Pennsylvania and Texas oils. The low gravity appears to indicate a radical difference between the constitution of the paraffin and the asphaltum oils.

Flash Point.—The flash point of California crude ranges from 400 degrees F. down to 60 degrees F. or lower. Most of the oil classed as fuel oil flashes about 150 degrees F., and practically all of it at 130 degrees F. It will be seen that none of the fuel oil of this State requires any preparation in regard to flash point to fit it for use. Considerable quantities of prepared fuel oil are now being put on the market. These oils being the refuse from distillation of lighters, the hydrocarbons

are entirely removed. This residuum or tarry substance is hard to burn, having a very high flash point, containing water and other foreign substances. Its specific gravity is about 14 degrees. It does not contain the thermal units of heat per pound of oil that a higher grade does, but its high flash point gives it a factor of safety quite necessary to be considered. All the flash points mentioned herein are by the open electric cup, a standard adopted by the California Oil Association.

TABLE I.

Field.	County.	Color.	Spec. gravity, at 60° F. Beaumé.	Flash, degrees F.	Burning point, degrees F.	Viscosity, 60°.	Viscosity, 185°.	Calorific values, in calories.
			Deg.					
Colusa.....	Colusa.....	Black.....	14.3	...	...	4.88	1.28	18.684
Napa.....	Napa.....	Red-brown	15.5	268	280	10.35	1.34	...
Humboldt...	Humboldt.....	Yellow.....	29.0	...	...	1.51	1.05	...
Humboldt...	Humboldt.....	Yellow.....	37.5	50	50	1.10	0.95	...
Humboldt...	Humboldt	Yellow.....	26.1	228	237	3.00	1.15	...
Humboldt...	Humboldt.....	Brown	48.6	63	65	1.17	0.96	...
Santa Clara..	Santa Clara....	Greenish...	34.4	...	...	...	...	20.146
San Meteo...	San Meteo.....	Brown	22.1	194	215	10.96	1.50	...
Colenga	Fresno.....	Brown	20.3	158	202	11.13	1.54	...
McKitterick	Kern.....	Black.....	17.7	...	...	63.15	2.12	18.675
McKitterick	Kern.....	Black.....	15.1	...	...	283.3	3.57	17.571
Sunset.....	Kern.....	Black.....	16.0	...	...	...	...	18.684
Kern River..	Kern.....	Black.....	13.4	...	...	...	...	18.473
Kern River..	Kern.....	Black.....	13.2	...	...	1,795.13	7.51	18.342
Kern River..	Kern.....	Black.	14.7	...	...	274.35	3.35	18.848
Kern River..	Kern.....	Black.....	16.0	...	...	273.11	3.67	18.646
Kern River..	Kern.....	Black.....	16.3	...	...	299.59	4.70	18.747
Summerland	Santa Barbara.	Black.....	12.1	...	...	...	8.35	18.998
Ventura	Ventura.....	Black.....	23.0	84	95	34.28	1.84	19.440

Calorific Value.—The calorific value of California crude oil ranges from 10,000 to 11,000 calories (18,000 to 19,000 B.T.U.) and averages 10,500 calories (18,900 B.T.U.). As a rule the higher the specific gravity the higher the calorific value, although this rule should not be applied too closely without a physical test. This average of 10,500 calories is 1.8 per cent. lower than the average of six of Pennsylvania's best pro-

ducing wells and 1.1 per cent lower than the average of six of Beaumont (Texas) wells (Pool's Calorific Power of Fuels, page 251). Taking account, however, of the difference of weight per unit of volume, the specific gravity of Pennsylvania oils being 0.84 equals 35 degrees Beaumé, of the Texas oils 0.9200 equals 22 degrees Beaumé, and of the California oils 0.9415 equals 19 degrees Beaumé. In Table I, following, will be found gravities, viscosities, flash and burning points and calorific values from a number of samples of California crude oils. The figures in this table are from a paper by Professor Edward O'Neil, of the University of California, published in the Journal of the American Chemical Society, July, 1903.

NOTES.

THE BATTLESHIP OF THE FUTURE.

By Rear Admiral R. H. S. Bacon, C. V. O., D. S. O., Associate.
Read at the Spring Meeting of the fifty-first session of the Institution of Naval Architects, March 16th, 1910. (Slightly condensed.)

This title may seem a bold one, and is, perhaps, misleading, for it is not the intention in this paper to attempt to foretell the design of battleships in the future; it is written rather with the idea of reviewing the chief features of battleship design and the direction in which they trend owing to progress in modern naval conditions, and so to start a discussion which may be of interest to this Institution.

The battleship is solely a gun platform. The gun is at the present moment the main weapon in deciding naval actions; therefore, progress in battleship design has continuously been in the direction of increased offensive gun power, and defensive armor protection against gun power. The development of gun power has followed closely upon the improvements in the manufacture of steel. The cast gun with spherical shot was superseded by coil guns, rifling, and pointed projectiles. The restriction of the chamber pressures which could be allowed in the early days involved low muzzle velocities and short range. Improvements in propellants and strength, and more reliable material for construction, have increased the length of range and led to longer guns and high pressures. The result has been that the accuracy of the gun of today at 12,000 yards is considerably greater than the accuracy of the gun of thirty years ago at 2,000 yards. At the present moment the development of the gun shows no signs of abating, and there is every reason to believe that, if required, the power of the gun can within the next few years be considerably in-

creased. The first important questions in design are: Is such an increase necessary? What advantages would it confer, and what disadvantages would it entail? To consider these we must examine the second feature of a battleship, namely, its defensive armor.

The distinctive feature of the battleship over the cruiser is that, in the battleship speed is sacrificed for defensive armor protection. This armor is disposed with its greatest thickness in the belt, gun positions and conning tower, so as to protect those portions which are vital, so far as maneuvering and gun fire are concerned.

There is no doubt that the gun of the present day can, with armor-piercing projectiles at a range of about 6,000 yards, penetrate any armor which can be practically mounted in a ship, and guns can be built which can penetrate this armor at ranges outside which it would be practically impossible to bring a fleet action to a decisive conclusion. Therefore, if anything approaching modern design of battleships is to be adhered to, it is perfectly hopeless to think of attempting to supply armor which would be impenetrable to any guns which may be constructed in the reasonable future. If such a thing were attempted, we would have to return to the shape and speed of the old *Inflexible*. Moreover, if such a ship were constructed, and impenetrable armor provided, the racking entailed by successive hits of heavy projectiles would cause such general leakage as to leave the ship little better off than if the armor had been penetrated.

This race between gun and armor, which has been going on for over half a century, is being decided for the moment in favor of the gun; nor is there any indication of there being a chance of improving the armor, and strengthening general construction so as to render ships reasonably immune from armor-piercing projectiles. So far, therefore, as regards mere perforation of armor with armor-piercing projectiles, there seems to be no immediate necessity for increasing the power of the gun.

The armor-piercing projectile, however, is not the most efficient shell for effecting general damage on a ship, and penetration of armor is not the sole object of gun fire. The chief damage will probably be done by heavy bursting charges. The heavier the bursting charge the greater the damage inflicted by each individual hit. There can be no choice, so far as damage is concerned, between a large burster and a small one, and therefore, from this point of view, the larger the gun and the larger the projectile the better. Again, accuracy in the performance of a gun can be best obtained by development of the gun and projectile in reasonable proportions. With power-worked mountings rate of fire need not be reduced as guns increase in size, and therefore the desirability of rapid hitting and effect of the hits when obtained both point to steady increase in the size of the guns. The sole limitation as regards progress in the size of gun carried is the increase in displacement necessary.

If the gun is to be increased in size on the same displacement, one of these three things must happen: (1) The number of guns must be reduced, or (2) the armor protection must be reduced, or (3) the speed must be reduced.

The extra weight required for the guns must be obtained from one of these three sources. We therefore find that while accuracy and efficiency point to increase in size of gun, this increase must be obtained at a sacrifice.

First, as regards displacement. It has often been said that our ships are too large, and as regards this there are two distinct issues: (1) Finance and expediency. (2) Efficiency and general utility.

The chief argument against size is the enormous cost of each ship, and, therefore, the large loss in case of sinking. Another argument is that our harbors are already too small, our docks require enlarging, and the ships become too large to handle in fleets. As regards the first of these arguments, the whole crux of the question seems to lie in the consideration as to whether a large ship or a small ship is more likely to be lost from the attack of a torpedo. Constructionally, there

seems no doubt that the larger the ship the more likely she is to survive the blow of a torpedo. Surviving, however, is not the sole point. Damage by a torpedo will temporarily incapaciate a ship however large she may be, and, therefore, a bigger money value of fighting appliance will be temporarily incapacitated by a single torpedo attack in the case of the large ship than in the case of the smaller; but the smaller ship may sink, and the loss become permanent. This question, therefore, resolves itself into one of time of repair. Provided, therefore, the number of ships are kept constant, although their size is increased, no danger from the torpedo accrues to the country by building larger ships. The matter, from this point of view, becomes one of the national purse, and whether, while retaining pro rata numbers, we can afford to build bigger ships.

There are, however, many who uphold the view that they prefer a larger amount of smaller ships to a less number of large ships, and that the money devoted to naval construction would be better spent upon numbers than upon size. The features of this contention require careful consideration. It must be assumed that an equal sum of money is spent in either case. The smaller sized ships involve carrying fewer guns, or a mixed armament of heavy and lighter guns, both of which are adverse factors to hitting at long ranges.

The most modern type of battleship at present constructed is represented by the *Dreadnought* class, and the improved succeeding vessels. Of course, it is well known that the chief features of the *Dreadnought* are the installation of an all-heavy caliber armament of the heaviest nature, accompanied by increase in speed. Whatever arguments have been raised against this type have been proved by subsequent experience to have little weight with European navies, since all have to a large extent adopted the principal features of the *Dreadnought*. It was only to be expected that the sudden transition from the *Lord Nelson*—or really, so far as general knowledge in the Navy and the country were concerned, the *King Edward* type—to the *Dreadnought* was certain to evoke criti-

cism and comment. The directness that characterized the Admiralty administration of that period swept aside all compromise, and designed a ship with no guns of any sort except the 12-inch and 12-pounder, the latter being then considered the most suitable for dealing with destroyers.

The main reasons underlying the abolition of the secondary armament were, first, that these guns tended to reduce the accuracy of hitting of the heavier guns; and secondly, that the weight, and therefore the extra tonnage that the installation demanded, could be better devoted to other purposes. The contention as regards accuracy of hitting has been well borne out by subsequent experience. Without entering into any technical reasons—and the argument is very highly technical —one point can be easily appreciated, and that is, that when the undivided attention of those directing the fire of the guns is concentrated on one type of gun only, it is far easier to correct the fire of the guns efficiently than to correct the vagaries of three calibers of guns, of varying rapidity of fire and of varying external ballistics, all of which are firing at the same object.

It must be appreciated that a 6-inch gun at long range is of no use at all against thick armor, and against the lighter armor and superstructures the small burster of the light weight of the shell militate against much real damage being done to the ship, even by a large number of such projectiles. It is held that "a hail" of such projectiles is liable to damage communications, etc. This might possibly have some weight provided that "the hail" really exists in practice, or that if it exists "the hail" hits the ship. As a matter of fact, when gun fire is handled so as to attempt to obtain hits, rapidity of fire falls very much below the possible rapidity of fire of the gun, and the actual rapidity of fire may easily fall to a quarter or less of the possible rapidity. In fact, when the fire of a quick-loading gun at long range is intermixed with other guns, rapidity of fire falls off very considerably, since it is now appreciated that rapidity of fire, unaccompanied by rapidity of hitting, is a futile waste of ammunition. It is generally

considered that 6-inch fire at ordinary battle range may be discounted when accompanied by 12-inch gun fire.

There are, however, a considerable number of people who hold the view that battles will not always be fought at long range, and therefore, in thick weather, the smaller guns may more nearly approach their theoretical value. It is perfectly possible that battles may be fought at moderate ranges, but it must not be forgotten that the accuracy and rate of fire of the heavier guns is also increased pro rata at such ranges, whereas the actual effect of the bursters and fragments of the projectiles at these ranges is not greater than at the longer ones, and that the damage occasioned by small projectiles is in no way comparable to those of the heavier type of guns.

If it were possible to install guns of this nature in a ship without either restricting the fire of the heavier guns or of unnecessarily increasing her displacement, and if it were thoroughly understood that they were only there to be used in thick weather, there might be some advantage in installing them. The difficulty, however, of such an installation is very great. If placed below the tier of heavy guns, their position comes very close to the water; if placed on a level, they are bound to reduce the arc of fire of the heavier guns; and if placed above them, they involve considerable top weight; since wherever they are put, if they are to be of value in a fleet or single ship action, they must be protected by adequate armor. The futility of installing any guns for use in an action unless they are adequately protected against gun fire is apparent. If the fighting power of the ship with a mixed armament is to be kept the same as that of a ship with a single-caliber armament, the small gun must be in addition to the large ones. The question of the installation of such guns is, therefore, purely one of whether the designers consider the extra tonnage to be worth the advantage that may accrue. Up to the present those responsible for the construction of our ships appear to take the view that the advantage is not worth the extra displacement.

The question of mounting an intermediate type, such as the

9.2-inch or 10-inch, as well as the heaviest type, is slightly different, since these guns have a projectile more capable of inflicting damage at long range; but when the disadvantage of a dual armament is considered, in conjunction with the comparatively small amount of weight saved, and the small increase in rate of fire obtained practically by mounting such guns, instead of an additional number of heavy guns, there is not much doubt that in the future the single heavy armament will carry the day.

There is one other aspect as regards the mounting of 6-inch guns, and that is from the point of view of those guns being used merely for anti-torpedo boat guns. This is advocated by those who look on the 6-inch gun as a better anti-torpedo boat gun than the 4-inch. This is purely a matter of opinion, and there is a greater excuse for mounting 6-inch guns for this purpose than for use in an action, since it does not involve the necessity of providing them with armor protection. Their use would then be largely confined to defence against torpedo craft before the day of a general action arrived, and for this purpose they require no armor protection. It is not likely that after a general action with the present-day heavy ordnance, these guns, whether protected or not, would survive to be of much practical value.

One point, however, must not be forgotten, and that is, that the possibility of mounting 6-inch guns in future ships sets a low limit on the thickness of armor belt, since it would obviously be foolish to reduce this to an amount which would allow these small guns to effect penetration at battle ranges, and therefore to lay the ship open to vital attack by small guns.

In building the small ship, speed, size of gun and power of each hit are sacrificed. Moreover, the length of the line of battle is considerably increased. The fewer the guns mounted in each ship, the larger the number of ships that will be required for an equal number of guns; therefore, the larger number of intervals between the ships, and an equivalent increase in the length of the fighting line. This is, further, a

very important consideration when the question of the use of the torpedo during an action comes to be considered.

As regards the size of our harbors, we have still plenty of places where we can berth ships 100 per cent. larger than the *Dreadnought,* and without real inconvenience. Size is by no means the only inconvenience in handling a ship in a close harbor, and from practical experience I can vouch that the *Dreadnought* was from every point of view more easily handled in a close harbor than the *Irresistible,* whose tonnage was 50 per cent. less. It must also be remembered that increase of size of ship in comparison with the size of harbor may be met by a development of methods and of auxiliaries such as tugs. As regards depth of water, although certain harbors may be less convenient than others, we have still plenty that can be used, even if greatly increased draught becomes a necessity.

As regards maneuvering with a fleet, no greater libel was ever published than the statement that the *Dreadnought* could not be maneuvered satisfactorily with the fleet, and any naval officer of experience who is not an automaton would find no difficulty in handling vessels of 50 per cent. greater tonnage, even when in company with the lighter class of battleships. The fact that the *Dido* class were about one-third the tonnage of the *Bulwark* did not prevent their safe handling when in company.

The use of the torpedo during a fleet action can no longer be neglected. The improvements of late years in this weapon have been considerable. It has been very much the fashion up to date to disregard the possible developments of this weapon. There is no doubt that of late years in America, on the Continent, and also at Messrs. Whitehead's, there have been introduced improvements into the torpedo which have totally altered the character and use of that weapon. It is not improbable that these improvements are merely in their infancy, and that before many years have elapsed we may find ourselves confronted with a weapon of considerably greater powers. If the range of the torpedo can be brought up to

6,000 yards it has great possibilities in an action. There is no doubt that it is far easier to hit one of twelve ships with a torpedo than to hit a single ship aimed at, and therefore the longer the line of ships, and the greater their numbers, the more vulnerable they will be to attack by destroyers at long range in an action.

This is an additional argument in favor of increasing the size of individual ships rather than the number of ships in the line.

As regards retention of the present thickness of armor protection, this is a matter which may before long undergo considerable modification, and the armor problem of the future appears to resolve itself into the answer to the following question:

Supposing the guns of the enemy can with certainty pierce armor protection at reasonable fighting range, what is the most economical thickness of armor to adopt? Very many considerations are involved in the answer to this question, such as the position and thickness of horizontal armor, but considering the enormous sacrifice in weight now made to carry thick armor protection, it is a matter that before long may undergo bold and radical revision.

We now come to the third possibility of saving weight, and that is, reduction in speed. Probably there was no point among the improvements introduced into the *Dreadnought* that was more violently attacked than the increase in speed from 19 to 21 knots, and the attempts made to prove that such an increase was of no value were legion. It is probable that this increase will in time be looked upon as one of the most valuable features of this design. It has been impossible for anyone to argue that, from a strategical point of view, such an increase in speed was not of value, but from a tactical point of view this increase was bitterly attacked. This attack arose from not taking a sufficiently broad view of the possibilities of the future.

The speed of a ship when she is built remains practically constant all her life. Judging from past experience it is im-

probable that the designed speed of a ship will ever be subsequently increased. Fifteen years ahead is a long time to look forward to, but the careful student of naval progress is forced to take into account the possibilities of all naval weapons. The improvements in the torpedo must in time introduce a class of tactics more analogous to single ship than fleet actions. It is in such a class of tactics that speed will become a valuable factor, and it is highly probable that the speed of battleships will increase up to that of the cruisers.

It is therefore probable that to get the maximum efficiency out of the gun and yet not to decrease the numbers mounted in a ship, increase in tonnage and reduction of belt armor of battleships will be features in future construction, and that for other reasons the value of speed will be enhanced.

It does not seem possible that the size of gun will be limited unless all nations agree to do so, and such an agreement is highly improbable. It would be a foolish policy on the part of any nation to install smaller guns than those carried in the ships of rival Powers. Again, it would be almost as unwise to provide guns of less damaging power than modern design permits. The number of guns carried in a ship cannot well be reduced below that of the *Dreadnought,* otherwise the difficulty of hitting at long range with such an armament will be considerably increased.

To sum up, all the considerations of offense and defense point to increase in size of battleships as modern gun construction advances. But, since the modern battleship no longer holds the supreme position which, in the old days, made the battleship the sole ultimate arbiter of sea power, it is improbable that, as the torpedo improves, battleships, unable to defend themselves against any form of torpedo craft, will be built merely to fight battleships. The functions of the large cruisers will therefore be assumed by the battleship, high speed will become more and more necessary, and armor protection will be less accentuated than at present. The link between the ocean-going destroyer and the battleship will become closer, and we may reasonably expect that the huge monsters

of the future will always be accompanied by torpedo craft of high sea-going speed as defensive and offensive satellites.

The battleship as now known will probably develop from a single ship into a battle unit, consisting of a large armored cruiser with attendant torpedo craft. Line of battle, as we now know it, will be radically modified, and the fleet action of the future will, in course of time, develop into an aggregation of duels between opposing battle units. The tactics of such units open up a vista of most exhilarating speculation, and will afford to the naval officer of the future a scope for his tactical skill never dreamed of by us or our predecessors.

The whole future is pregnant with radical obliteration of our present notions as regards tactics, but we may confindently prophesy that size of ships and power of gun will increase and increase until war, the great arbitrator among theories, will confirm or reconstitute our opinions regarding naval armaments.—"The Engineer."

THE DREADNOUGHT CONTROVERSY.

The paper on the battleship of the future read by Admiral Bacon at the Naval Architects last week revived in a mild form the controversy which raged round the design of the *Dreadnought* some time ago. Sir Gerard Noel accurately described the paper as a defense of the *Dreadnought,* and we may treat Admiral Bacon's rejoinder that no defense is necessary much as a judge would a counsel who submitted he has no case to answer. Now, as every schoolboy knows, the features which distinguished the *Dreadnought* from all previous battleships were principally the increase of dimensions and speed and the absence of a secondary armament. In the words of Admiral Bacon: "The directness which characterized the Admiralty administration of that period swept aside all compromise and designed a ship with no guns of any sort except the 12-inch and the 12-pounder." The increase of dimensions, coupled with the introduction of turbines, made possible

the increase of speed, so that it is not necessary to refer to dimensions and speed separately. The wisdom or otherwise of the increase of size affords little scope for profitable discussion, since in the history of naval construction there are two outstanding indisputable features—that size always tends to increase, and that from any one stage in the development no return is possible to smaller dimensions. Once the step is made it is irrevocable, and even those who regret the increased cost of the *Dreadnought* type will agree that unless some discovery of a revolutionary character is made a return to smaller battleships is out of the question. Vessels of corresponding or larger size are being built by foreign Powers, and it would be folly to build on a smaller scale than our neighbors.

The real point in this discussion of size is often missed. It is the only profitable point to consider in this part of the controversy, and it is this: Displacement will grow as time goes on, but it ought to follow a natural law, and not be artificially hastened. It would be as reasonable to argue that since we must all "shuffle off this mortal coil" we may as well commit harri-kari at once, as rapidly to increase the size of our ships because we foresee that some day the displacement will be double what it is at present. This country, above all others, requires a numerous fleet, and as our financial resources are not infinite, increase in cost of individual ships must eventually result in a restriction of the number, and it would appear fairly obvious that, subject to the individual vessels comparing favorably with contemporaneous foreign vessels, they should be as small as possible. The point is, we repeat, not whether it is possible to get greater fighting powers on a larger displacement. If that were the only thing to be considered the resources of engineering would allow us to go up to vessels of 50,000 or 60,000 tons immediately. What we have to consider is whether the development of foreign vessels calls for increase of size, and, if so, what is the minimum which will give us the necessary advantage. The balance of evidence is against the contention that the increase in the *Dreadnought* was at the time justified by the action of foreign Powers, but

as we cannot now go back it ought to be a cardinal consideration that similar steps are not taken in future. Unfortunately, the *Dreadnought,* instead of, as was hoped, stifling competition, has put additional life into it, and we have already reached in this country a displacement of 26,500 tons, while in America battleships of 32,000 tons are in contemplation. The situation is analogous to the wars of rates which occasionally disturb the commercial world, and which, if they do not ruin one or more of the competitors, are only ended by an agreement to limit prices. It is too much to expect, in view of the non-success of efforts in this direction in the past, that any agreement is possible towards limitation of armaments, either in respect to size of vessel or numbers.

We turn to the other feature of the *Dreadnought* design, and here, as Sir William White remarked, it is astonishing that every considerable naval Power except ourselves remains convinced as to the value of a powerful secondary armament. A distinction must be drawn between the smaller guns, such as the 12-pounder or 4-inch, which are only expected to be of service against torpedo attack, and the secondary armament proper, which, strictly speaking, should embrace any guns of caliber between the main armament and the 6-inch weapon. Some people, indeed, consider the 6-inch as an ideal anti-torpedo gun, though it would be almost as accurate to describe a sledge hammer as the ideal instrument for cracking nuts. Torpedo attacks are not made in the open when the vessel attacked has ample time to train her guns on the torpedo boat or destroyer, unless it is at the end of an engagement against a practically disabled ship. They are made under cover of darkness or some other favorable screen, and to be effective the attack must be sudden and unexpected. To meet such an attack under these conditions volume and rapidity of fire rather than strength of individual hits is essential, for it is fairly certain that much of the shooting will, even with the best-trained gunners, be wide of the mark under such adverse conditions. The numbers of guns mounted for this purpose is therefore more important than their size, providing

they be of such caliber that a single shot can disable the adversary. A single 12-pounder shell will wreck a destroyer's engine room, and a 4-inch shell accurately placed between wind and water would in all probability sink her. As the effective range of the torpedo is increased the size of the anti-torpedo gun must go up also, so as to be equally effective against destroyers at the longer ranges, but it is easy to see that the smaller the gun which is suitable for the work the greater will be the number of positions which can be found for mounting it in a vessel of given dimensions. These are considerations more or less outside the range of controversy, but the efficacy and desirability of mounting guns intermediate in size between the anti-torpedo and main armament is a question on which opinion is very sharply divided in this country, though, as we have stated, other navies seem to entertain no doubts at all on the subject. The reasons which Admiral Bacon gives against the secondary armament are two in number. The first is that these guns tend to reduce the accuracy of fire of the heavier guns, and the second that the weight, and therefore the extra tonnage, that the installation demand could be better devoted to other purposes. He also stated that they could not be mounted without restricting the fire of the heavy guns, but this argument was demolished by Sir William White, who pointed out that in some American ships eight large guns had been employed with a powerful secondary armament free from the defect which Admiral Bacon said must accompany it. As to the question of fire control it does not require an expert to appreciate Admiral Bacon's point as to the difficulty of correcting the fire of different calibers of guns with varying rapidity of fire and varying external ballistics. The matter is no doubt of importance with respect to gunnery returns and target practice, but the point is what exactly will be the extent of this disability under real conditions. The experiments carried out against the old battleship *Hero* are reported to have shown that fire-control stations are not likely to survive after the first few minutes of an

engagement, and if this is the case firing would become independent, and each gun's crew would have to rely upon its own judgment. Under such conditions some accuracy of fire due to the absence of fire control would be lost, but it does not appear likely that any one gun would interfere with another's accuracy.

No one can read the absorbing narrative by Captain Semenoff of the battle of Tsu-Shima without being struck by the undoubted havoc which the 6-inch guns of the Japanese vessels caused, and the fact that since the war the Japanese themselves have laid down battleships with powerful secondary armaments shows that they appreciated the part which these guns played. Admiral Bacon, in his reply, endeavored to discount the lessons of Tsu-Shima by the arguments that this battle was fought with old-type ships, and that gunnery had enormously improved since that date. From a less powerful advocate than Admiral Bacon these arguments would appear entirely inconclusive. We must assume equality of type of ship on either side, and what was true of battleships of the *Royal Sovereign* and *Majestic* era should also be true, *mutatis mutandis,* of vessels of the *Dreadnought* era. The argument based on improvement of gunnery also has little weight unless we assume that the fire control remains undamaged, and so gives the single-caliber ship the advantage. Another point, which escaped notice at the meeting, is the fact that larger guns have a much shorter life than the smaller. The longer a decisive action is deferred in a naval war the more important relatively becomes the secondary armament, as the big guns become badly worn and accuracy of shooting falls off more rapidly than in the case of the smaller guns. This point is brought out in Captain Semenoff's book, and no doubt in a measure accounts for the large proportionate effect produced by the Japanese 6-inch guns. The question of secondary armament is, as Sir William White said, a vital one, and in view of the decided course taken by other navies should be carefully reconsidered.—"The Engineer," March 25, 1910.

ELECTRICITY ON WARSHIPS.

The application of electricity has its limitation, or, as it was teresely put by a well-known engineer recently, electricity is not like a patent medicine, it will not do everything. In no case is this more true than in its use on board naval vessels where, though the advantages in certain functions are very great, its limitations are none the less well defined. It is necessary to insist on this truth, because we have recently seen in practice and proposal that in the natural development of the application of electricity there is a tendency rather to shut the eyes to its disadvantages in certain particulars and to apply it without a full measure of justification. It is, of course, human to "magnify our office," and the electrical engineer is no less human than the rest of us. Neither should proposals for the extension of electrical appliances be cold-shouldered or discouraged, since it must be admitted that the limitations to which we have referred are by no means final. On the other hand, any new proposal must have strong presumptive evidence in its favor that it will accomplish what has hitherto been performed by other means in a simpler and more efficient manner.

Take the case of the stokehold fans fitted in all warships. Here was a case of exceptional difficulty for the steam engine so long as it was of the open type. The speed of revolution was necessarily high, and the difficulty of keeping the bearings in adjustment, owing to the presence of coal dust, was well nigh heartbreaking to the engineer officer in charge. If they did not break down more frequently it was because their use was restricted by the small proportion of steaming done at high power, at other times they were run at slow speed, if required, merely to ventilate the stokeholds. The application of the enclosed engine with forced lubrication changed all this, but though the difficulties were thus overcome a large number of ships was fitted with fans driven by electric motors. The policy has now been abandoned and a return made to steam-driven fans, not because the electric-driven fans were

unsatisfactory but because they did not accomplish what the steam-driven fans do in a simpler and more efficient manner. We should have thought this would have been apparent from the first. Here we have a boiler requiring to be supplied with air by a fan in close proximity to it. What could be simpler than to take a steam pipe from the boiler, or steam pipe in the boiler room, to a steam engine coupled to the fan, and what more satisfactory, providing the engine is reliable? The alternative by electric drive is to take the steam hundreds of feet in some cases from the boiler to an electric-generating engine, bring the transformed energy back to the boiler room by electric cables, and apply it to the fan by a motor fixed in many cases in dust and heat, which are almost as fatal to prolonged efficiency, even in the best-designed and protected motors, as the coal dust alone was to the open-type steam engine. There is also an objection from an administrative standpoint. All electrical apparatus in his Majesty's ships is placed under the torpedo officer, and in the case of the electrically-driven forced draft fan we have the curious anomaly that the engineer officer who is responsible that the vessel's machinery is kept in constant readiness to develop its full power when required, is dependent upon another department for the air pressure, without which he is unable to maintain the full power. We need not labor the point; every business man is alive to the danger to efficiency by dividing responsibility. It has been an unfortunate feature of the policy of past Admiralty Boards to endeavor to minimize the growing importance of engineering by assigning parts of the engineering work of the ship to others than the engineer officer, who, by training and experience, is most fitted to perform it, and to call the work so detached by another name. The considerations to which weight should be given in the application of electrical and other forms of transmission on board warships are protection and suitability for the work required. While for machinery operating within the limits of the engine and boiler rooms, and below the protective deck steam is the simplest, most direct, and most suitable, there is much to be gained by confining the use

of steam machinery within those limits. Steam pipes are bulkier and more likely to be damaged by fragments of shell than electric cables, while from the point of convenience of leading and absence of heat, the latter are much to be preferred where it is necessary to take one or other through the living spaces or magazines. For ventilating fans outside the engine and boiler rooms, for fire and other pumps not in machinery spaces, and for some other auxiliary machinery, the substitution of electric motors for steam engines is a gain in comfort, convenience, economy and efficiency. When, however, the more important auxiliary engines outside the machinery spaces come to be considered the question of the suitability to the work to be performed becomes of vital importance. The most important of these engines are for working the capstan, the helm and the guns. Most vessels have now an electrically-worked after capstan where the work is not of an exceptionally heavy nature, but for the forward capstan, where the load is heavy and often fluctuating within wide limits in the course of a few revolutions of the capstan, the electric motor is not so adaptable, and steam is still generally preferred. It is generally understood also that there would be no saving in weight by the electrically-driven capstan, over the steam driven. So much excellent work has, however, been done by electrical engineers and manufacturers in the application to winches and the like that there is little doubt that the forward capstans in our warships will in time be electrically driven. In the case of the steering engine the main problem is to arrange an electrical device which shall perform the same function as that performed by the hunting gear on the steam engine, so that the movement of the helm shall correspond exactly with that given to the wheel by the steersman. In the steam engine this gear is extremely simple and reliable, and practically the only danger of its getting out of order is in the long mechanical connection between the wheel on the bridge and the valve on the engine. The problem of electric hunting gear has been worked out and applied in at least one merchant vessel; but it is only right that in our warships so important an auxiliary,

on which to a great extent the safety of the vessel depends, should be treated with the greatest conservatism. In any change the burden of proof lies on the innovator, and there must be strong presumptive evidence, which must appeal to an impartial judge and not merely to those specially interested in the particular form of the change, that the new application will accomplish all that the old one did, and perform it, moreover, in a better and more efficient manner.

If we apply this principle to the question of working the heavy guns we shall be at a loss to understand why the experiment of working the turrets of the *Invincible* electrically was embarked upon. We commented some weeks ago on the fact that this vessel has done very little service since she first commissioned in March last, and that her main armament had not yet been made effective. We notice in a service contemporary that Lord Charles Beresford, in a speech at Portsmouth, is reported to have stated that the 12-inch guns cannot be worked, and that if a war broke out the first thing that would be done would be to turn over her complement to another ship. This is a serious statement, which we can only hope is greatly exaggerated. At the same time it appears that an error of judgment has been made in trying electrical gear on so large a scale, especially at the present time when early completion of all our vessels is greatly to be desired. We do not profess to know the nature of the difficulties which have been encountered—we can only surmise by a consideration of the magnitude of the change involved from hydraulic to electrical. The hydraulic gear, which has been brought to such a state of perfection, is admirably suited for such work. It allows of the greatest precision of movement; it contains within itself its own brake by virtue of the enormously increased resistance to the movement of the water as the speed is increased; it is practically "dead beat," and by virtue of the very large torque and relatively slow movement of the hydraulic motors, it is essentially suited to the starting and bringing to rest the very large masses comprised in the gun with its mounting and shield. The electric motor in its native simplicity has none of these things. In the matter of the last-men-

toned quality, it is the antithesis of the hydraulic motor, as its characteristic is relatively high speed and small torque, and this must involve greater complexity of gearing. The mechanical difficulties to be overcome on this point alone must have been considerable. Whatever the difficulties—and they cannot be insuperable, although we can hardly doubt that the final result will not compare favorably with the hydraulic system—we trust they will be speedily overcome. The case of this vessel is one that points the moral of the exercise of a sense of proportion in engineering matters, and emphasizes the remark we quoted at the beginning of this article: Electricity is not a patent medicine.

SUPERHEATING.

For some time past the steam-engine problem has been attacked at both ends. Superheaters and condensers have each undergone development. Vacuums have been insisted on, sought for, and attained which no one wanted for reciprocating engines. On the other hand, the temperature of steam has been raised. In the best steam engine a gas, not a vapor, is the working fluid. The first steps in either superheating or condensation are easily taken. As we proceed our difficulties increase, our troubles augment, and finally the engineer begins to ask himself whether it is worth while to go further. The moment the answer in the negative is given progress is at an end. The utmost limit of economy has been reached, and some other motor than the steam engine must be found to supply more economical power. Obviously, then it is a matter of much interest to settle what the limiting conditions are. It would take up too much space to deal with both superheating and condensation. We shall confine our attention to the first.

In a recent impression we dealt with the theory—or, to be more comprehensive, the theories—of superheating. It is unnecessary to go over the ground again. The practical fact made abundantly clear by experimental inquiries carried out

at various times during the last sixty years or so is that superheated steam is more economical than saturated or normal steam. How much more remains to be seen. It is probable, indeed, that everything depends on the conditions under which it works in the steam engine. It may be that an engine is wastefully using wet steam. No change is made save that superheating is introduced, with, as a result, a saving of 30 per cent. If, however, the engine was previously economical, working with steam from which all the water was as far as possible removed, then the saving might not exceed 10 per cent. It is necessary to insist on this view, because there are so many expressions of opinion, mostly based on results obtained under different conditions. We are told, on the one hand, that it is not worth having, and on the other that it is simply invaluable. It is by no means easy to establish any standard of value under the circumstances. This much may be assumed, however: Given any degree of superheating which will secure the presence—initially—of dry steam in the cylinder, and some economy will be secured; but the percentage will follow a curve, showing that the total saving will, expressed in percentages, continually fall. To put this into figures, for the sake of illustration: If the saving for 100 degrees F. was 20 per cent., for 200 degrees it would probably not exceed 25 per cent., or for 500 degrees 30 per cent. In writing this we know that we express opinions to which some of our readers may take exception; yet while, as a matter of course we do not bind ourself to precise figures, we believe that when we say that the economy secured falls off much faster than the temperature rises, we only state that which has been proved in practice. We have, however, before us a report on certain experiments carried out in the United States with the steam yacht *Idalia* to test the value of superheated steam, which apparently go to show that the percentage is uniform. She has four-cylinder triple-expansion engines, with cylinders 11½ inches + 19 inches + 22 1/16 inches + 22 1/16 inches × 18 inches, and piston valves. Steam is supplied by Babcock & Wilcox boilers, burning anthracite, with 65

square feet of grate and 2,500 square feet of tube surface. She has a superheater consisting of horizontal U pipes with 340 square feet of surface. The engines indicate about 500 horsepower. The highest superheat was 105 degrees F., the lowest 57 degrees F. The general results show a net gain of 10 per cent. with the maximum rise. The pressure at the throttle valve varied between 190 pounds and 203 pounds. The steam consumption was about 16 pounds per indicated horsepower per hour. Lieutenant Halligan, U. S. N., who carried out the trial says: "The tests indicate that there is a net gain of 10 per cent. obtained by using 100 degrees F. of superheat. This may be expressed as a saving of coal of 1 per cent. for each 10 degrees of superheat." Now, from the latter conclusion we dissent, that is to say if it is intended to have a general application. It is, however, quite true of the *Idalia,* the curve of economy being virtually a straight line. It is a pity that it could not be extended by raising the degree of superheat. The benefit derived from moderate superheating will depend largely on the quality of the steam; the dryer that is the smaller the saving.

When we raise the temperature to the highest point compatible with the endurance of the engine, we get into a different region. The saving now is not solely due to the drying of the steam, but to that and the prevention of cylinder condensation as well; and it may reach as much as 20 per cent. even when applied to highly economical engines. So far we have been dealing with steam only, and the weight used per horsepower per hour. We have now to consider the total cost. The higher the superheat, other things being equal, the larger must be the superheater; and it must be kept constantly in mind that the transfer of heat from one hot gas to another through an iron plate is a slow and difficult operation. If we want very hot steam we must have a great deal of surface exposed to gas or radiant heat at a very high temperature. Furthermore, to get good results the steam ought to move at a high velocity for reasons well understood. This somewhat complicates the design. Abroad separately fired superheaters

are favored. If the superheater is placed in the boiler flues the feed-water heaters, or "economizers," are likely to suffer. The more moderate the rise in temperature required the smaller and cheaper will be the superheater, and it is in settling the conditions which must make for all-round economy that the skill of the engineer will be manifested.

A good deal of literature of a singularly unconvincing kind exists concerning superheating. There remains a wide field for inquiry. At present we find two types of advocates—the one is all for moderate temperatures and the maintenance of conditions favorable to the prolonged existence of the superheater and the engine. The other party energetically demand the use of very high temperatures. They maintain that even though the wear and tear of the superheater may be great, yet that it can always pay for itself in a few months; while as for the engine, with piston valves, hard metal and heavy lubricating oils, or graphite and water, no risk of cutting rubbing surfaces will be incurred. So far it is impossible for anyone to pronounce a dispassionate valuable opinion, because there is not enough knowledge. Although superheating is little used for British ships, about 250 German and French steamers have been fitted with superheaters. These ought to supply a great deal of information; but none has been made available save the general statement that an economy of 20 per cent. in fuel has been secured. But half a century ago as much as 25 per cent. was obtained in Cunard ships, and yet superheating did not pay. Progress has been made since; no doubt increased durability has been secured, and superheating has apparently been put on a sound commercial footing, but definite statements of facts are lacking. It seems certain that a careful collection of information from those who are using superheating, extending over the last half-dozen years, would be of value. There are, as we have said, many reports published from time to time, but they seldom deal with a period of use sufficiently extended; and they for the most part omit essential information. We are told, for example, that Mr. X. has fitted a "Will-o'-the-Wisp" superheater to his engine, and secured

an economy of 16 per cent.; but not a word of accurate information is vouchsafed as to what the previous condition of the engine was, the cost of the installation, its effect on the boiler efficiency, or what the cost of upkeep might be. Anomalous as the words sound, the superheating of steam is an old new thing, and much remains to be learned about it that both engineers and steam users would do well to know. It is good, but just how good or when best remains to be settled.—"The Engineer."

TWO, THREE OR FOUR PROPELLERS?

Considerable diversity of opinion exists with regard to the respective merits of twin, triple or quadruple propellers when considered from the point of view of efficiency alone. This is not to be wondered at, for the problem presents rarely, if ever, a simple issue. It is generally confused in ship designs, by conditions which modify the result, and prevent a direct comparison on the basis of propellers pure and simple being made.

In coming to a definite conclusion as to the relative efficiencies to be expected when adopting one, two, three or four screw propellers to drive a ship, great care must be taken to allow for all the factors involved. These may be roughly divided into two groups—those that deal with the motive power inside the ship and those that deal with the actual application of that power to propulsion outside of it. There are causes within the vessel which sometimes of themselves demand a multiplicity of propellers, such, for instance, as the application of a large total indicated horsepower, while keeping individual units within practical limits. The exigencies of design of expansion turbine engines also create a necessity for three or four shafts, in order that the engine room may be reasonable in length. In warships the importance of keeping the height of reciprocating engines with the limits demanded to obtain adequate shelter below the protective deck leads to the same position. With these it is not our purpose to deal here, but rather with the actual causes which may be expected to modify

the efficiency of propulsion when the number of propellers has been already fixed upon.

The work, which has to be done by the propulsive agent, is not only to overcome the bare resistance to motion of the ship form, or, as Froude called it, the "tow-rope resistance," but also to overcome a considerable augmentation of that resistance, caused by the diversion of the natural stream lines of the hull, by the propellers. This is a very variable factor, and has been given by different investigators as amounting to 45 or 50 per cent. of the tow-rope resistance in extreme cases, while falling as low as 4 or 5 per cent. in others—obviously a serious consideration.

With twin screws, another item which must be added to the total thrust to be exerted, is the unavoidable increase of resistance due to the shaft supports. This also occurs in the case of three or four screws, but in varying amount. Thus the actual resistance to be dealt with is dependent upon the number of propellers employed, even where the design of hull proper is unchanged. In addition to this there are two items which occur when more than one propeller is used, namely, the influence on efficiency of different directions of turning, and also the effect which the propellers have on one another, in absolute restriction of the free flow of water to any particular propeller of the set.

The incidence and magnitude of these varying factors form a problem of considerable complexity, and it is not an easy matter to generalize in such a way as to cover particular cases.

The battle of single against twin screws has been fought and won, with honors easy as far as efficiency goes, where the conditions of draught are ample for the single screw. Sir William H. White championed the cause of twin screws as against single screws long ago, even where the draught was unresricted.

The wake of a ship may be considered as composed of two elements: the frictional wake—that is, the water to which forward motion is imparted by the surface friction of the hull—and the actual stream-line motions due to closing in of

the water along the run of the ship. While the single screw revolving close to the ship may utilize some portion of the energy in the frictional wake, and thus gain in effective thrust, it also, by virtue of its proximity, affects the stream lines more than do twin screws, and thus produces a greater amount of augmentation of resistance. Against this must be set the fact that the twin screws have to overcome the additional resistance due to the shaft supports. As a general conclusion, with supports of good design, and where the diameter of propellers is ample, it may be assumed that there is little to choose from an efficiency point of view simply, between these two propositions. In a great many cases it is, however, probable that the single screw suffers, by comparison with twin screws, from the fact that the diameter is too small for the thrust which has to be developed, and hence erroneous conclusions are drawn. Considerations of immersion for all conditions may lead to this practice, and obviously efficiency at sea may be adversely affected where the propeller tips break the surface owing to pitching of the vessel, and so produce air drawing.

From this point of view alone twin screws have a decided advantage, because they are not only of smaller diameter, but, being usually fitted further forward than a single screw, they are not so much affected by pitching. It has been noted also that emersion of the tips of twin screws due to rolling is not a serious item, for the ship usually rolls with the wave, and the screws retain their depth more or less. We may then assume that for at least a majority of cases the plea for twin screws rests on a strong foundation.

Much depends, of course, upon the additional considerations of shaft-support design and of direction of turning. In these matters both experience and experiment have led to an almost uniform practice in modern shipbuilding, namely, that the best way for turning twin screws is that the propellers revolve their tips outward from the ship at the top when going ahead, and that the shaft supports be extended right out to the propellers where possible. This avoids the necessity for propeller struts, or "A" frames, and care is of course taken

in design that the inclination of the shafting is not set downward at the after end, sufficiently to make a heavy drag from this cause at the bossing.

The crux of the whole matter is in the next step we have to consider, and to which designers are often driven, namely, the adoption of triple screws. What has already been said of single and twin screws obviously applies to a system which is a combination of both; but the problem is further complicated by the mutual interaction of the three propellers upon one another. The best direction of turning, too, is more difficult to decide upon than in the case of twin screws, although the usual practice is to turn the side screws as in twin-screw installations, and the center one the same way as the starboard. That this is the best way is by no means sure, for examples of different combinations existing in the Navy fail to demonstrate conclusively the advantage of one arrangement over the other for all cases.

Admiral Melville, Engineer-in-Chief of the United States Navy, read a paper in 1899 before the Institution of Naval Architects, which dealt with the triple-screw United States cruisers *Columbia* and *Minneapolis*, in which he made great claims for the improved efficiencies obtained with them, as compared with twin-screw cruisers. He estimated that for ships of from 12 to 20 knots speed, there was a gain of 8 per cent. by using twin screws as against single screws. He further estimated that for triple screws as against single, there was a gain of 5 per cent. for 15-knot ships, rising to 12 per cent. for 20-knot ships. Unfortunately these figures were more nearly assertions, than actual proofs, and perhaps the best comment upon them is an extract from Sir W. H. White's contribution to the discussion upon the paper. He said: "I have great difficulty in dealing with the paper, as it contains a great many contentions, not a few speculations, some arguments and very few facts." Sir William considered, and wisely, we think, that Admiral Melville's conclusions rested on a somewhat slender basis. This is an old story now, but facts

are unchangeable, and recent experience does not lead one to modify the general purport of it to any great extent.

Modern developments in large power marine installations, the adoption of Parsons turbines, or in war vessels fitted with reciprocating engines, the necessity of keeping cylinder heads below the protective deck, make it sometimes inevitable that the transmission be divided into three or more units, and so we are face to face with the problem, whether we will or not.

Now these cases are not necessarily identical. Division of a large total power derived from reciprocating engines will almost certainly involve the fitting of three propellers of fairly large diameter, and this will bring the tips into close juxtaposition. The same thing may apply when triple screws are adopted on account of restricted draught of water. When the reason for triple screws, however, is the use of expansion turbines, the revolutions are much higher, and the diameter of propeller consequently much less, in order to obtain a moderate pitch ratio. The element of interference with one another may thus not act with equal intensity in each case, owing to difference in clearance of tips. We must therefore remember this in considering the two cases. Now it is well known that good efficiencies have been and are attained with turbine steamers having triple screws, but it has certainly not been demonstrated that triple screws are better than twin screws for all cases. On the contrary, as we shall see, the evidence is all the other way. The triple system is simply unavoidable in some cases, and the lesser efficiency must be accepted if it exists. In the case of turbine propellers of small diameter the issue is further obscured, because there is sometimes a sacrifice of efficiency due to this cause alone. We must look, then, for some facts upon which to base our judgment.

It has often happened that the French engineer in matters of pure science or experimental research has shown the way to the world, and anticipated the march of events. This question of twin and triple screws is a case in point, and we make no apology for quoting freely from the very valuable facts

available, which have a much wider application today than they had when ascertained and put on record.

The following abstract, written nearly a quarter of a century ago, might very well have been penned yesterday: "When the attempt is made to substitute three or four screws for a single one, or for twin screws, the question is—not to find out if the efficiency is greater from the point of view of speed in proportion to power, but simply to examine if efficiency be not too much sacrificed. The object of increasing the number of propellers is to obtain certain qualities, but not to realize an increased efficiency. It is not sufficient, moreover, to make sure that the use of several screws is possible; the best relative positions when three or four propellers are employed must also be studied, in order to do away, as far as possible, with the action of mutual interference."

Here, then, we have the problem set by French marine engineers so long ago, and stated in exact terms the inquiry which is still being made, but with much more insistence now owing to the increased necessity for use of multiple propellers.

A committee were appointed by the French Ministers of Marine to control, analyze and report on the experiments to be made. A launch was constructed at L'Orient to the designs of M. L. de Bussy, Inspecteur General du Génie Maritime. This launch represented on a scale of one-tenth of full size, an ironclad, 344.5 feet long, having a displacement of 9,600 tons. It was fitted with three fast-running steam engines which drove the launch at speeds corresponding to 17.22 knots for the ship. The *Carpe*, as she was named, represented very well then modern up-to-date practice, in spite of the fact that the hull was not quite the shape that modern practice dictates. This was, however, a common factor to the whole of the investigations.

It is interesting to note that the committee put on record the fact that these miniature trial trips could only be carried out in calm weather, because the least ripple of the water was the same to the *Carpe* as a rough sea to the vessel she represented. Care was taken to count the revolutions and to record the

power over a measured distance; to eliminate tide influences, keep the hull clean and to prevent permanent list of the vessel. All this is significant in the light of the importance attached to recent experiments made with a large model of the *Mauretania*, constructed by Swan, Hunter & Wigham Richardson, Limited.

All reasonable precautions seem to have been taken, and the committee set themselves to elucidate the principal questions, which would be raised by the use of a system of propulsion with three screws, and particularly to decide between two different positions of the side screws, viz: one position in the same transverse plane as the center screw, and the other situated more in advance, the shafts being also brought closer together. The speeds were taken to vary directly, and the revolutions to vary inversely as the square root of the dimensions. For 17 knots speed the revolutions were 120 per minute for the ship, so that we see the condition does not correspond with turbine practice, but more nearly with the triple-screw reciprocating-engine system. The side screws were turned outward on top in the usual present-day manner.

Here we have a valuable contribution to the subject. The first series of experiments was made with propellers having the reasonable pitch ratio of 1.25, all in the same transverse plane; then with three others in the same position, having pitch ratio of 1.55; thirdly, with two large propellers to represent twin screws, the pitch ratio being again 1.25. It may be assumed, for purposes of comparison, that the propulsive areas were identical for all three sets. In the first series of experiments, it was found that the triple screws with the larger pitch ratio, were better than triples with small pitch ratio, and that neither of them was so good as the twin screws of the same disc area. The question of pitch ratio is a little foreign to our present consideration, but is valuable and interesting to note. More than this was, however, observed, and we lay stress on the following point:

During the tests with triple screws it was found that "an increase of speed was gained when the central screw was sup-

pressed." Further tests confirmed this result; we shall allude to this striking fact again.

A second series of experiments was made with the side screws advanced an amount corresponding to about 10 feet for the ship, forward of the position of the center screw, each being brought about 1 foot nearer—for the ship—to the middle line, at the same time. Two changes were thus made simultaneously here, not a desirable procedure, but perhaps permissible in the circumstances. In this position the triple screws of 1.25 pitch ratio were slightly better than those of 1.55 pitch ratio, and better even than twin screws in the forward position. This forward position is, of course, neither suitable nor necessary for twin screws.

The results, then, may be summed up as given by the committee: "Three screws are, from the point of view of speed, very nearly equivalent to two screws of the same propulsive surface, and immersed to the same depth, when the most favorable position is chosen for each system."

From an efficiency point of view the twin screws have the advantage generally of both single and triple screws in these experiments. How are we to account for this, especially in the light of good efficiencies having been obtained by some turbine steamers?

Let us consider the first controlling factor, interference. Here again we need not go away from the report of the same commission. From experiments made with a small propeller of about 2½-inch diameter, driven at 540 revolutions per minute in still air, they deduced the following result: A screw propeller draws air from all points situated in front of a transverse plane passing through the extremity of its blades, and even a little from behind this plane; it then projects this air in a truncated cone, or more correctly, a sort of truncated pear, the small base of which is a circle equal to the diameter of the screw; the generating line being inclined on the outside at an angle of 30 degrees to the axis. Properly speaking, the surface generated by this line is a zone of unstable equilibrium, within which the flame used for test purposes is repelled, and

on the outside of which the tendency to attraction is manifest. It is important to remark that the screw in this case is stationary, and if we had to do with a vessel in motion, the truncated cone would be drawn out so as to approach the shape of a cylinder of water. The deduction they make from all this is as follows: "In short, one may conclude from this experiment that if the three screws interfere with each other at great speeds when they are placed in the same transverse plane, it is not because they send water to each other in consequence of the phenomenon designated by the name of dispersion, but rather because they dispute with each other for the possession of the water they draw upon." Exactly, and we think no one could put the matter more aptly, more concisely, or with more truth at the present day.

Before inquiring into the other factor, of the amount of augmentation of resistance caused by side and center screws, it is interesting to note that some experiments were made upon Italian torpedo cruisers of the *Tripoli* and *Folgore* class, fitted with one, two and three screws. The results are not strictly comparable, being taken at varying speeds, the center screw also being low down, and placed far aft of the ship. A broad comparison can, however, be made, and it brings out the fact that twin screws were better than triple screws, and about equal to single. This is valuable as strengthening the foregoing deductions made from a model, by experiments on full-sized ships.

Reverting now to the curious fact recorded in the first series of experiments on the *Carpe*, that "increased speed resulted from suppression of the center screw." Is this borne out in any experience with full-sized ships? Curiously enough, the publication of the *Carpe* results brought out the fact that three-screw steamers of comparatively shallow draught and fitted with triple screws, the centers of propellers being all in one transverse plane, had given some extraordinary results on trial. The vessels were 98 feet long, with 18 feet beam, having screws of 2 feet 9 inches diameter. When they were tried, they attained a speed of 10 miles per hour.

Now comes the astonishing result, which was given by the builders, that when the center screw was stopped there was no appreciable dimunition of speed.

The ship attained the same speed with two engines as with three engines working, notwithstanding, it must be assumed, that there was the drag of the center propeller to overcome. One infers from this that the ships must have had very full coefficients. The reason assigned by the builders for this result was "interference" of the screws with one another, and this they proposed to cure by fitting each screw with a drum and shield, attached to the ship. We may be permitted to conjecture that very little, if any, improvement would have been attained. What, then, is the explanation? The center screw, when being driven, is absorbing power from the engine, and must therefore be exerting thrust which ought to appear as increased speed of ship; but it does not! What, then, has become of it? The center screw, in its struggle to obtain a supply of water, finds its two companion screws acting as competitors instead of coadjutors, and consequently has to deflect the stream lines over a wider area than it should do otherwise. The augmentation of resistance is increased so much by the center screw in these extreme cases as to balance the increased thrust, and thus leave the speed unchanged. This seems the only explanation which accounts for the fact that the center engine is expending power which must produce a corresponding thrust at the center screw, specially in view of the frictional wake, and yet there is no increase of speed attained.

We may therefore conclude that any triple-screw system will be subject to this serious drawback to a greater or less degree, depending on the proximity of the propellers to one another and to the hull of the ship, and thus we should deduce that three screws could never be better than two, other things being equal. This presupposes, of course, that each system is properly placed and has ample disc area.

The investigation of results with three propellers leads on naturally to the consideration of the efficiency to be obtained when using four. The problem is similar, though we have not

to consider a center screw, which is probably the worst position in a multiple system.

We may suppose, too, that four small screws for a turbine installation will have a better chance of obtaining a clear run of water than four larger ones suitable for a sub-divided reciprocating installation. The shaft tubes will be smaller, and a saving of resistance effected.

From experiments made with models of the *Lusitania* and *Mauretania* by John Brown & Co., Limited, and Swan, Hunter & Wigham Richardson, Limited, the direction of rotation of quadruple propellers to give the best efficiency was found to be that the after screws should turn outward on top, and the forward ones inward on top when going ahead, and this arrangement was adopted in the ships. The published propulsive efficiency of the *Lusitania* was only 48 per cent. at the top speed, this being the relation of E.H.P. to shaft horsepower, and the *Dreadnought* class are reported to give about the same efficiency.

This is not a very high attainment, and although propellers of large area have been fitted, and better speed results obtained since with the new Cunarders, we are not able to say how much, if any, of the improvement is due to better stoking and consequently increased shaft horsepower. Failing this, we must assume the efficiency to be not greater than 50 per cent. It was currently reported also that the *Mauretania* obtained as good a result with three propellers as with four, after losing one by accident.

This, if true, might also be due to increased transmission of power on the other three shafts, more steam being available for the turbines, but it is curious in the light of the circumstances we have been considering. On the whole there seems no reason to doubt that most of the disadvantages that accrue to a system of triple screws, in regard simply to efficiency, obtain in a greater or less degree with a quadruple system. This brings us to the final conclusion, that twin screws hold the field still, where adequate area of disc and moderate pitch ratio can be obtained, with sufficient immer-

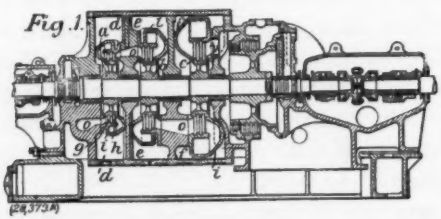
sion of tips to ensure a supply of solid water—to use a sailor's paradox. Other conditions than efficiency, pure and simple, are left out of account.

No stress has been laid upon the possibility of cavitation affecting one system more than another, it being assumed for this purpose that care is taken in each to keep well clear of the breaking-down condition for the propellers.—"The Engineer."

TURBINES—C. A. PARSONS' LATE PATENTS.

December 29, 1908.—This invention relates to partial-admission fluid-pressure turbines of the type in which the velocity of the working fluid is maintained chiefly or entirely by dropping pressure, small amounts at a time, in guide-blades only, a sector of such guides being arranged in front of each ring of rotating blades. A high-pressure turbine is divided into wheel elements *a, b, c* within closed chambers *d, e, f*. Each wheel carries a plurality of rows of crescent-shaped blades *g* and *h*. Opposite to the steam inlet *o* to each chamber

Fig. 1.



is a casting *i*, which carries the first sector of guides *j* in front of the first row of rotating blades *g*, a similar sector of guides *m* being carried in front of the second row of rotating blades *h*. These guide-sectors form the first steam-way. After the steam has been passed by these guide-sectors once through the two rotating rows of blades, it is delivered into a passage which directs it in a curved path radially outwards; a passage *l* then directs the steam over the ends of the running blades and radially inwards to a space at the front end of the wheel.

From this space the steam passes in the same direction as before, through two more sectors of guides j^1, m^1, alternating with the blades g and h, these sectors forming the second steamway, which discharges the steam through the opening q freely into the chamber. To the partition-wall opposite to that which carries the casting i there is secured another substantially similar casting r, which carries two sectors of guides j^2, m^2, alternating with the same rotating blades g and h, and forming a steamway, through which the working fluid from the chamber e passes in the same direction as before, into a space s; the steam is then directed radially outwards and over

Fig 2.

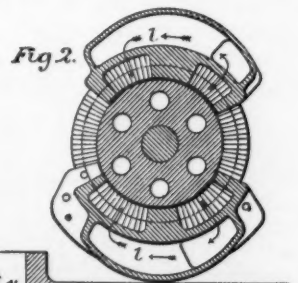
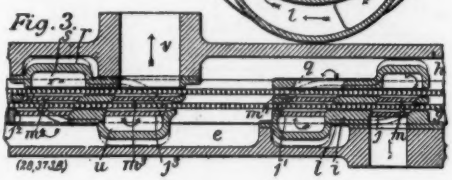


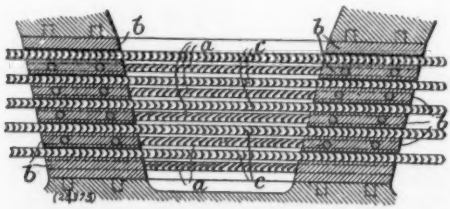
Fig. 3.



the ends of the blades by a passage similar to the passage l in the other casting to a space u, whence the steam passes through the sectors of guides j^3, m^3, and the same rows of rotating blades g and h. The steam leaving this fourth passage-way is discharged by the passage v in the partition wall into the first steamway in the next chamber, where it is similarly dealt with.

December 29, 1908.—This invention consists in a partial-admission turbine of the type in which the velocity of the working fluid is maintained chiefly or entirely by dropping pressure, small amounts at a time, in guide-blades only, and in

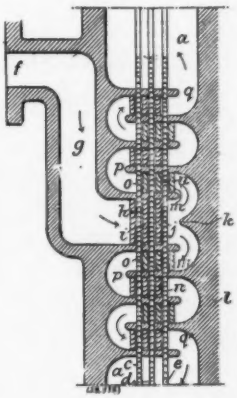
which no expansion takes place in nozzles preliminary to that occurring in the guide-blades, a sector of guide-blades being arranged in front of each ring of rotating blades, divided into a number of separate turbine elements, each having a rotor working in a separate chamber, and each element having rotating blades *c* alternating with segmental guide-blades *a*, which terminate in short end blockings *b*, the drop in pressure per element being so small that the amount of expansion of the fluid in any chamber shall not exceed about four volumes. According to the invention, each turbine element is provided with a sufficiently small number of rows of blades to reduce the loss arising from spilling of the steam at the ends of the segmental guides to a small quantity. Moreover, the pressure existing in any chamber is approximately a mean between that of the fluid entering and that of the fluid leaving the chamber.



From the foregoing it will be understood that small drops of pressure take place only in the guide-blades, the rotating blades being of a form suitable for abstraction of velocity only. If the steam be passed only once through the rotating blades, a plurality of rows of rotating blades are employed; but if the steam is passed a plurality of times in the same direction through the same rotating blading, a single row of blades per wheel may be employed, always providing, however, that the expansions and drop of pressure per wheel is slight, as above indicated.

December 29, 1908.—This invention relates to partial-admission fluid-pressure turbines of the type in which the velocity of the working fluid is maintained chiefly or entirely by dropping the pressure, small amounts at a time, in the guides,

a sector of such guides being arranged in front of each ring of rotating blades. Within chamber *a* a turbine wheel carrying one or more rings of working blades—say three rings *c, d, e*—is enclosed, the steam is led from the inlet *f* through a passage *g* to a sector of guide blades *h*, so formed as to produce a slight drop of pressure and to increase the velocity of the steam passing through them. This steam impinges upon the first ring of rotating blades *c*, which are of crescent shape, of symmetrical form. After leaving the blades *c*, the steam passes into another similar sector *i*, which again slightly decreases the pressure, and restores the velocity. This stream again impinges on a row of crescent-shaped or symmetrical blades *d*, in which



no pressure drop occurs, and then passes through another similar sector *f*, which deals with the steam in the same way as before, delivering it at increased velocity to the third row of working blades *e*, which are of crescent or symmetrical form. The steam passing out of the last row of blades *e* impinges against a wedge buttress *k* on the opposite wall *l* of the chamber; this buttress divides the steam into two streams and reverses the direction of axial flow, and also causes half of the steam to flow to the left of the main passage-way, and half to the right—*i. e.*, in opposite circumferential directions. This reversed steam then passes through sectors *m, n, o*, similar

to those previously described, and through the same rows of blades *c*, *d*, *e*, which alternate with the sectors, and is discharged into a curved passage *p*, which once more reverses the axial direction of flow of the steam and directs it again through the same three rows of rotating blades and through alternating sectors in front of them. This reversal of axial flow may be repeated several times, and then the steam is delivered through the passage *q* directly into the chamber *a*. It is essential, however, that the amount of expansion which the steam undergoes in passing through and through the rings of blades shall be but slight.—"Engineering."

IMPURITIES CAUSING SCALE AND CORROSION.*

The chemist has shown the way in which to prevent scale and corrosion in boilers and also how to prevent losses in the industrial arts. His method is to remove from the water the objectionable salts which it contains by changing the soluble salts into insoluble precipitates, which can then be removed by sedimentation and filtration before the water is used. This process is rational in application, the results certain, and the cost in every case is but a small fraction of the advantage gained.

Natural water supplies furnish the water converted into steam; these supplies are rarely, if ever, pure, for water in its descent to the earth as rain absorbs carbonic acid, some air and other impurities. The carbonic acid absorbed enables it to dissolve certain salts of lime and magnesia. Other substances will be dissolved, depending upon the nature of the rocks, soil, vegetation, sewage and industrial waste with which it may come into contact.

Steam generation is a continuous process, fresh feed water being supplied to the boiler as the water evaporated into steam

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*"General Characteristics of Salts, Gases and Acids Which Cause Scale or Corrosion in Boilers. Density of Water and Its Purification." Abstract of paper read before the American Institute of Chemical Engineers, by J. C. WILLIAM GRETH.

leaves it; this results in a continual concentration in the boiler of the impurities introduced with the feed water, since none but volatile impurities pass out with the steam. The nonvolatile impurities collecting in the boiler manifest themselves as suspended matter, scale, corrosion, or by an increased density of the boiler water.

The suspended matter may be carried in with the feed, or may be due to substances forced out of solution as a result of either heat or concentration, or both. Scale formation in the boiler is due to the action of heat, pressure and concentration on the impurities in solution and suspension in the feed water. Corrosion of the boiler is due to the introduction of gases and acids, or their formation from some of the impurities in solution in the feed water, by the reactions resulting from heat, pressure and concentration. The increased density of the boiler water is due to the concentration of the sodium salts and of the scale-forming salts, to the limit of solubility.

Scale is the great bugbear which steam users, as a rule, fear, and make more or less of an effort to combat, and with good reason. Scale is one of the crucial items entering into boiler-operating costs. Scale can nearly always be attributed to the lime and magnesia salts in solution in the water. The character of the scale depends on the acids combined with the lime and magnesia; on the type of boiler in use, and on the rate, temperature and pressure at which the boiler is operated. For instance, the carbonates of lime and magnesia, when present alone, usually form a soft scale. The presence of calcium sulphate sometimes increases its hardness. A calcium-sulphate scale is generally quite hard.

The following are a few of the items which, from an economic standpoint, make it almost imperative to prevent scale formation, or at least to remove it periodically:

First. Reduced evaporation due to the insulating effect of the scale on the heating surfaces of the boiler.

Second. Cost of labor required for cleaning the boilers and auxiliaries.

Third. Cost of repairs to boilers, necessitated by their being

subjected to over-heating on account of the heating surfaces being scaled.

Fourth. Loss of efficiency and earning power of improved furnaces and stokers installed to increase evaporation, which correspondingly increases the concentration of impurities, thus forming a greater deposit of scale, and hence a greater reduction in the efficiency and life of the boilers.

Fifth. Cost of tube-cleaning machines, repairs to them, interest and depreciation on money invested, and labor and power required for operating them.

Sixth. Cost of boiler compounds, or any substances introduced into the boiler to prevent the adherence of the scale-forming matter to the shells and tubes.

Seventh. Loss due to the investment in spare boilers to be put into commission when it is necessary to take boilers out of service for cleaning or repairs.

Eighth. Waste of fuel due to heat lost in cooling a boiler for cleaning or repairs, and that required to bring it to steam again.

Ninth. Loss due to reduced efficiency of boiler auxiliaries, especially in the feed-water heaters and economizers, resulting in lower temperatures of feed water, thus materially increasing fuel consumption.

Salts Which Enter Into Scale Formation.

Calcium Carbonate.—This salt is in solution in natural waters as the bicarbonate. On heating the water carbonic acid is driven off and the normal carbonate is precipitated to the limit of its solubility, which in distilled water is about two grains per U. S. gallon, but in waters containing other salts at boiler temperatures and pressures it varies from about one to five grains per U. S. gallon. This limit of solubility remains almost constant for a particular water under boiler-operating conditions. The precipitation of calcium carbonate by heat is practically complete at about 300 degrees F. The precipitation, however, starts as soon as the temperature of the water is raised and continues until the limit is reached. The precipi-

tation therefore occurs, not instantaneously, but gradually, and with a diminution of precipitate as the limit of solubility is approached. This is true of all scale-forming salts that are precipitated by heat alone.

The amount of calcium carbonate left in solution in the water depends upon the other salts in solution. Heat alone will effect the removal of both the free and the half-bound carbonic acid; therefore calcium carbonate will be precipitated, and the precipitate may eventually deposit as scale. The formation of scale from precipitated calcium carbonate depends upon the other substances in solution and the conditions under which the boiler is operated. For instance, if the water contains sodium carbonate, the chances are that the calcium carbonate will be precipitated as sludge. If, on the other hand, the water contains calcium sulphate, the cementing action of the calcium sulphate will tend to form a hard scale, the hardness of which will depend upon the amount of calcium sulphate in solution in the water, and the rate, temperature and pressure under which the boiler operates.

Magnesium Carbonate.—This substance has the same general characteristics as calcium carbonate, being held in solution as the bicarbonate. The normal magnesium carbonate, however, is more soluble than the normal calcium carbonate. Further, magnesium carbonate is quite easily dissociated as a result of heat, liberating carbonic acid and precipitating magnesium hydrate, which, at all temperatures, is very insoluble, rarely over one-half grain per U. S. gallon. The analysis of boiler blowoff waters will usually show both magnesium carbonate and magnesium hydrate in solution, while the scale will generally show magnesium hydrate.

Calcium Sulphate.—This sulphate is soluble in natural waters to over 100 grains per U. S. gallon, and under boiler temperatures and pressures to approximately 25 grains per U. S. gallon, depending upon the other salts in solution. It is quite generally stated that calcium sulphate is insoluble at 300 degrees F.; this may be the case in a solution of calcium sulphate in distilled water, but it is not the case with natural

water supplies or those containing other salts in solution. The analyses of hundreds of samples of blowoff waters show calcium sulphate present to the extent of 25 grains, where temperatures far above 300 degrees F. are maintained. The amount held in solution at boiler temperatures depends upon the amount of other substances in solution, and also upon the rate of concentration of those impurities. Calcium sulphate generally gives a hard scale, deposited in layers. This is probably explained as follows:

In the boiler the calcium sulphate concentrates until it forms a supersaturated solution, from which, on agitation of some sort, it quickly deposits a mass of densely interlacing hard crystals of gypsum, until its concentration drops to the point of saturation. Further, concentration in the boiler again forms the superheated solution, from which later another crystallization occurs. These repeated periodic crystallizations of white gypsum, separated by the slow, constant and regular deposition of other scale, would give the laminated appearance generally seen in a calcium-sulphate scale.

Magnesium Sulphate.—This substance at boiler temperatures is quite soluble, and when present alone is not likely to form scale, but in the presence of calcium carbonate will react with it, forming magnesium carbonate and calcium sulphate. Magnesium sulphate is also objectionable because it reacts with sodium chloride, forming the very soluble sodium sulphate and magnesium chloride. This reaction is the result of heat and concentration in the boiler.

Calcium Chloride.—This lime salt is very soluble at all temperatures, its solubility increasing with the temperature. It is, however, a fact that with the increase of calcium chloride as a result of concentration, a point is reached where the calcium chloride begins to be dissociated, forming calcium hydrate and hydrochloric acid. The calcium hydrate is quite insoluble at boiler temperatures. Analyses of scale and sludge from boilers fed with water containing much calcium chloride show calcium hydrate, and evidences of corrosion, no doubt due to hydrochloric acid, are usually found. The calcium hydrate formed

as a result of this reaction may combine with carbonic acid, either introduced with the feed water or that liberated as a result of heat, and form calcium carbonate.

Magnesium Chloride.—This chloride has the same general characteristics as calcium chloride, except that it is more easily dissociated, and whenever present scale and corrosion result. The scale is due to the magnesium hydrate precipitated, and the corrosion to the hydrochloric acid liberated. In waters containing calcium carbonate the hydrochloric acid thus formed may be neutralized by the calcium carbonate, forming the calcium chloride and liberating carbonic acid.

Calcium and Magnesium Nitrates.—These salts have the same general characteristics as the calcium and magnesium chlorides, but the quantities present in most feed waters are usually so small that not much consideration is given to them. However, there are some water supplies in which these salts are present to such an extent as to cause both scale and corrosion.

Silica.—The silica in solution in a water usually does not exceed two or three grains per U. S. gallon, and by itself will not form scale, but it is always found in the scale when present in the feed water. Silica under boiler temperatures may react with sodium chloride, forming a sodium silicate and liberating hydrochloric acid.

Oxides of Iron and Alumina.—These are not usually present to any great extent, but by concentration enter into the formation of scale.

Organic Matter.—The substances included under this general term play an important part in the formation of scale, and in many cases when present cause the formation of a hard scale which otherwise might be quite soft. The reverse is also true with some forms of organic matter. Then, again, some organic matter may prevent the formation of scale by increasing the solubility of some of the lime and magnesia salts.

Sodium Salts.—These are present in nearly all water supplies and cannot be classed as scale-forming matter, although present to a slight extent in nearly all scale; this is due to their

being present in solution in the water, mechanically held by the scale, rather than from being forced out of solution, for long before such saturation is reached it becomes impossible to operate the boiler.

Corrosion.—Corrosion is the most dangerous of the various troubles due to impure feed water, and the one in many cases the most difficult to overcome. It is usually due to the acids introduced into the boiler in the feed water, or those formed as a result of reactions between various substances in solution, caused by heat and concentration; in some cases it is due to the oxygen of dissolved air. The different acids cause different kinds of corrosion, and it occurs in different parts of the boiler, depending upon the nature of the acid.

Gases and Acids Entering Boiler and Causing Corrosion.

Oxygen.—Nearly all waters contain more or less oxygen dissolved from the air. The oxygen of the air is more soluble than the nitrogen, and is frequently the cause of pitting and grooving, especially in those parts of the boiler where the temperature is low and the circulation slow, such as, for instance, in the mud drums of those types of boiler in which the mud drum is not in the direct path of circulation. The corrosion by oxygen is the direct formation of the various oxides of iron. It is next to impossible to overcome this form of corrosion, as there is no practical way of removing the dissolved oxygen from the water. The corrosion, however, is of a mild form and does not, except in rare cases, cause much trouble. The cases where trouble from this source is experienced can almost always be charged to the design of the boiler, for if the circulation is as rapid as it should be and all of the boiler water moving, the oxygen will go off with the steam, possibly causing some corrosion at or above the water line, but not generally to an appreciable extent.

Carbonic Acid.—This acid, which is mixed with the air to the extent of about 0.04 of 1 per cent., is present in all natural water supplies. It is absorbed by the water, in which it is quite

soluble, from the air. The corrosion caused by carbonic acid is usually indicated by pitting and grooving, and it is shown not only in the water space of the boiler, but also above the water line and in steam lines. Its corrosive action is much greater when oxygen is present with it.

Hydrochloric Acid.—This acid is rarely if ever present in natural waters, but is formed as a result of the decomposition of some of the chlorides, and reactions between such substances as magnesium sulphate and sodium chloride, and attacks surfaces exposed to steam. Hydrochloric acid is volatile, also very soluble; therefore corrosion is shown both above and below the water line. This acid attacks the iron, forming the unstable iron chloride which, at boiler temperatures, is dissociated; the iron being precipitated as the oxide, or hydrate, and the acid thus liberated combines with more iron from the boiler and is in turn again set free. In this way the corrosion goes on indefinitely, constantly increasing in its action on account of the continued increase in the acid contents of the water by the acid that is liberated from the impurities in the water being added to the acid formed by the decomposition of the iron chloride. This concentration of the acid does not take place above the water line, but the heat there is sufficient to dissociate the iron chloride. Corrosion from this cause is usually shown by pitting and grooving, rather than over the entire surface.

Sulphuric Acid.—When this acid is present in a natural water supply it is due to drainage from coal mines or industrial plants. It is not volatile and its action shows only below the water line. Its action on the iron of the boiler is similar to that of hydrochloric acid, except that it forms the iron sulphate, which in turn is dissociated into sulphuric acid and the iron oxide or hydrate. This iron oxide usually forms a part of the scale, or is present in the water as suspended matter, giving to the water the characteristic red color of iron rust. A feed water containing only a small amount of sulphuric acid will produce active corrosion, resulting in the destruction of the boiler, on account of the continual formation of iron sul-

phate and its dissociation into sulphuric acid and iron oxide or hydrate. Many water supplies, especially those contaminated with the waste from galvanizing plants, contain iron sulphate, which, under boiler temperatures, is immediately dissociated.

Organic Acids.—Under this head are included acids such as tannic and acetic. They are usually the result of contamination from vegetable or organic matter. The corrosion from organic acids is comparatively mild, but occurs to a greater or less extent, and is very similar to that from the other acids. However, the amount of such acids present in most waters is usually so small that little attention need be paid to it.

Density of Water in Boilers.

The increase in density of the water in the boiler cannot be prevented, for the evaporation of water into steam leaves the sodium salts in solution; and there is no means by which these salts can be removed from the water, either before or after it enters the boiler. By frequent blowing off the concentration of the sodium salts in the water in the boiler can be reduced, but not entirely prevented.

That portion of the scale-forming salts soluble at boiler temperatures and pressures also increases the density of the water, but these salts are constantly concentrating and precipitating, so that after a certain point is reached for uniform pressure and rate of operation, the analysis of boiler water will remain practically the same, with the exception of a variation in the calcium sulphate and an increase in the sodium salts.

Scale and corrosion are closely related, because of the number of salts which, as a result of heat and concentration, either decompose or react, forming salts and liberating acids; the precipitated salts forming scale and the acids causing corrosion.

The analysis of the water is of undoubted value in determining the substances in solution. There is, however, among chemists a wide difference of opinion as to the proper method of making combinations from the determinations of the various substances in solution. Experience enables a chemist to

formulate certain rules, and by careful observation during the course of the analysis, to note the salts present in a particular water. But in reporting the nature of the possible scale formed by a certain water, or the corrosion which might result from its use, not only the analysis of the water must be taken into consideration, but the reactions between the various salts in solution; these reactions, however, do not take place to the same extent in all waters. The amount of scale-forming impurity in the feed water rarely if ever bears a direct relation to the substances in solution in the water after concentration in the boiler, but it does to the amount of scale or sludge formed. However, there is a close relation between the amount of sodium salts introduced with the feed water and the amount found in the boiler water after concentration; this ratio indicating approximately the number of concentrations.

It cannot be definitely foretold that in a certain water containing both magnesium sulphate and sodium chloride there will be a reaction between these salts, yet hundreds of blowoff analyses show the results of these reactions, and the boilers show corrosion resulting from the liberated hydrochloric acid.

It therefore means a careful study of the water and the conditions under which the boiler operates, to determine whether scale or corrosion would result from the use of a certain water. It is almost impossible to predetermine the nature of scale from the analysis of the water. The only safe way is to feed water into the boilers, free from those substances which scale and corrode. Such general statements that waters containing only the carbonates of lime and magnesia will form a comparatively soft scale, and that the calcium sulphate will form a hard scale, and further, that it will increase the hardness of the carbonate scale, should be made with caution, for there are hundreds of instances where a hard scale is formed from waters containing only the carbonates of lime and magnesia, and also where the scale is quite soft in the presence of considerable calcium sulphate.

The nature and amount of scale formed in a boiler depend largely on the rate at which the boiler operates. For instance,

in some boiler plants operating considerably below their rating, and fed with water containing as high as 30 grains of both carbonate and sulphate scale-forming salts, in a given time comparatively little scale is formed, and that quite soft; while in others, where the water contains only about 10 grains of these same salts, and the boilers are worked above rating for the same time, a considerable deposit of hard, tenacious scale is formed. The type of boiler also has a bearing on the hardness of the scale. The scale in the water-tube boiler is generally harder from the same water than that formed in the return-tubular boiler, or in the old two-flue boiler.

Softening and Purifying Water.

To soften and purify a water properly means, primarily, a properly designed apparatus in which are met the requirements for complete chemical reaction. These may be summed up as follows:

1. An accurate chemical treatment, accomplished by the introduction of the proper reagents in exact quantities to react with the impurities in a definite quantity of water.

2. Thorough mixture of the reagents with the water to insure complete chemical reaction.

3. An accelerated chemical reaction, brought about by a thorough mixture of reagents and water, and by mixing the sludge of previous softening with the new finely divided precipitate. Heat will hasten the reactions, but is not essential.

4. A complete chemical reaction, brought about by a thorough mixture of the reagents with the water and by having the apparatus large enough to allow sufficient time for all the reactions to take place, and the apparatus so designed that every part of it is effective.

5. A rapid sedimentation, by having the new finely divided precipitate weighted by the sludge of previous precipitation, to cause it to settle more rapidly and perfectly.

6. A perfect clarification, by allowing time for sedimentation and final clarification by perfect filtration.

The proper softening and purification of water is, in a sense, a delicate operation, notwithstanding the large quantity of water usually handled. It is not merely a matter of lime and soda ash, but the intelligent use of the proper reagents to bring about softening and purification for a particular water supply, with neither an insufficiency of reagents nor too great an excess. A water containing 30 grains per U. S. gallon of scale-forming matter is harder than the average, yet in percentage this means only 0.05 of 1 per cent. of scale-forming impurity. Such a water completely softened should not contain more than three grains of scale-forming matter, or in percentage only 0.005 of 1 per cent. When these facts are considered, some idea is obtained of the accuracy of the treatment required for completely softening water. Of course, any reduction of the scale-forming salts is an advantage, but the maximum reduction can usually be obtained for very little extra expense with a properly designed apparatus, when such apparatus is given the necessary attention.

If a water supply contains less than four grains of lime and magnesia salts, but contains suspended matter, it should be clarified by sedimentation and filtration. If the water contains more than four grains of scale-forming salts, it should be softened and purified, that is, the reduction of the soluble impurities (not including the sodium salts, which cannot be removed) to a point where an analysis will show quantities about as follows: Volatile and organic matter, one grain; silica, one-half grain; oxides of iron and alumina, trace; calcium carbonate, two grains; magnesium hydrate, one-half grain; but no other compounds of lime and magnesia. Suspended matter should never be more than a trace. Such a water will not form scale nor cause corrosion. It will not form scale because the amount of scale-forming salts left in solution is too small, even with concentration, to form anything but a light sludge. This sludge can be kept at a minimum by proper blowing off, and the boiler, no matter how long it is in operation, will on being opened have the appearance of having been whitewashed; the iron of the boiler can be exposed anywhere by rubbing with

the finger or washing out with a good pressure. Corrosion cannot take place because the water is slightly alkaline and does not contain either corrosive acids or salts which, by dissociation or reaction, will form corrosive acids.—"Power and The Engineer."

KEROSENE IN BOILERS.

Kerosene is not listed as a scale solvent, and an analysis of the feed water is absolutely necessary in order to treat it successfully. By this I don't mean to say that kerosene is of no use in keeping boilers clean, because if that were so, there has been a large quantity of good oil wasted. Most everybody uses it as a cure for rust, on other iron, therefore I don't see why it won't cure the same thing, although called corrosion, on the inside of a boiler.

We used to put kerosene in our boilers in homeopathic doses. But we started to sell steam for heating purposes and the customers complained of the odor, so we discontinued the practice. We now use a compound made to fit our case.

First in the list of the impurities that cause scale or incrustation are sediment, mud and clay. The remedy for these is filtration or blowing; and let me add here that sometimes one can blow down until the operation costs more money than a filtration plant, and not get as good results.

Second come bicarbonate of lime, iron and magnesia scale. The remedy is a good heater, adding caustic soda, lime or magnesia.

Third, take sulphate-of-lime scale; add carbonate of soda or barium chloride. Be careful not to get in too much soda or it will cause foaming; the addition of barium chloride will prevent this.

Fourth, sulphate of magnesia will cause rust or corrosion; the remedy is carbonate of soda.

Fifth, mine waters contain acids and will cause corrosion; alkali should be added.

Sixth, the dissolved carbonic acid that the editor says is a

good thing to have in the smokestack will cause corrosion in the boiler when combined with the life-giver, oxygen. The remedy is a good heater, caustic soda and slaked lime.

Seventh, oil, animal or vegetable, will cause corrosion. We generally get it from the engine exhaust. The remedy is slaked lime and filtering carbonate of soda or kerosene oil. Here is where kerosene is good; and in no other place.

Eighth, sewage causes foaming and corrosion. The remedy is to treat it with alum or ferrous chloride, and filtering.

A word of advice: If one starts to use a compound to take off the scale and prevent foaming, do not wait a month before opening the boiler or boilers to apply the treatment.—ALFRED GRISWOLD, in "The Power and The Engineer."

Having read a great deal pro and con regarding the use of kerosene oil in boilers, and having noticed that very little has been said about the reason for success or failure, an explanation of the action of kerosene on scale may prove useful as well as interesting.

Some scale that forms in boilers is more or less porous, and is, therefore, capable of absorbing liquids like water or kerosene. If the boiler is wet, the pores in the scale will probably be full of water; and if oil is put in, very little will be absorbed. If the boiler is dry when the oil is put in, the scale will absorb considerable quantities of the kerosene. The ideal way of getting the oil into the scale would be to throw it around on the dry scale inside the boiler until the scale had absorbed all it would hold. But this cannot always be done, so the next best way is to run some oil into the cold, dry boiler and gradually add water until full. The oil floats on the water and reaches the scale before the water does. Consequently much of the oil will be absorbed along with the water, and the slower the water is run in the more oil will be absorbed.

Water, when heated, gives off steam. In the same way, kerosene oil, when heated, gives off a gas. For the poorer

grades of oil enough gas is given off, when the oil is heated to 110 degrees F., to burn in the air if lighted with a match; for the better grade of oil the temperature for this "flash point" will be up toward 150 degrees F.

Now, the temperature of boiling water is 212 degrees F., and as the pressure in the boiler increases the temperature gets much higher. Suppose we feed kerosene into the boiler when getting up steam. Then the oil quickly forms a gas and passes out with the steam. Very little, if any, ever touches the scale, and consequently we can't expect it to affect the scale.

When the oil is put in the wet boiler and floats on the water when getting up steam much the same thing happens, as very little oil every really gets into the scale. But suppose the oil has had a chance to soak into the dry scale and fill every pore before commencing to steam up. Then as the boiler gets hot some of the oil becomes a gas; and, as it gets hotter, more gas is formed. This gas needs more room than the oil occupied, so it expands and pushes out of the little pores and breaks the particles of scale apart in its effort to get free.

When the boiler is cooled down and opened up we would expect to find the scale all cracked loose, and this is exactly what happens. The scale must be taken from the boiler at this time, or the blanket of scale on the bottom may allow the metal to become too hot and "bagging" may result.

Therefore, the essential thing is to get the oil directly into contact with the scale. The oil is not a solvent by any means—it merely acts in the pores of the scales like dynamite in a hole in a rock when blasting.

Hence, we see that kerosene fed into the boiler with the feed water cannot prevent scale forming, nor remove any already there. But oil that really gets into the scale and is then heated loosens the scale by cracking. This explains why some engineers claim great value for oil as a boiler cleaner, and others, just as honestly, claim it to be worthless. Think it over and then try it for yourself.—L. P. HORSFALL, in "Power and The Engineer."

BOILER CORROSION.

One of the most dangerous faults which may develop on board a steamship, after a certain amount of wear and tear, is the corrosion of the boilers, and a considerable amount of close attention and inspection is required in order to safeguard the boat against danger from this cause. Boiler corrosion is due to three main causes—air, acid and galvanic action. Of these the two former, air and acid, enter the boiler from the outside with the feed water, while galvanic action is set up inside the boiler by the heat and either alkaline or acid water acting on the different metals employed in the boiler.

Corrosion, however formed, shows itself in pitting and eaten surfaces, and it is usually found at points where the heat impinges on the boiler to the greatest extent. These surfaces are generally along the lines of the fire bars on the water side of the furnaces, on the sides, bottom and middle of backs of combustion chambers, also in a lesser degree on the combustion-chamber stays nearest to the combustion chamber, and sometimes on the main stays nearest to the uptake end of the boiler. All of these parts should be very carefully examined when a boiler is laid off for repairs or while in dock. On the external surface of the boiler corrosion is caused by rust and by any burning of the heated surfaces which may take place. There is another kind of corrosion, technically called "grooving," which is sometimes found at the seams of the boiler, and is due to the mechanical action on the metal owing to expansions and contractions of the boiler.

In order to avoid the various types of corrosion mentioned above it is necessary to adopt certain precautions and remedies, which may be detailed as follows: It should be reckoned as an axiom that air should be excluded from a boiler as much as possible, as it tends to rust and corrode the plates. A most useful means of performing this is by the use of feed heaters, in order to thoroughly heat the water before entering the boiler. This not only drives off a large amount of contained air, but also, by sending the water into the boiler in a hot condition, tends to

keep up a better circulation in the interior of the boiler; bad circulation is a very potent cause of corrosion of boilers, inasmuch as there is always a certain amount of oxygen in the water, which is liberated by heat. If a boiler has bad circulation this free oxygen attacks the metal plates and forms rust; whereas in a boiler possessing good circulation the free oxygen is detached from any surface upon which it may happen to rest and rises to the steam space.

In order to safeguard against exposed metal surfaces forming a favorable opportunity for rusting, new boilers should be lime-washed in the water and steam spaces before using, and if rust is found on the plates after the first running the lime-washing should be repeated. In the meantime a scale should be induced on the boiler plates by keeping the density low by means of scumming, and also by the use of lime added to the feed water. This, of course, should only be used in moderate quantity, as only a very thin scale is required in order to protect the interior of the boiler from rust.

It will be easily understood that acids attack boilers very readily, and these are brought into the boiler by the feed water. The chief source of this acidity is the decomposition of oil, which arises from a too frequent internal lubrication of the engine. Where lubrication is necessary only the best mineral oil should be used for the pistons and piston rods. Moreover, the gland packing should preferably be greased by mineral oil rather than with the tallow which is so frequently employed. Animal and vegetable oils always decompose in the high temperature of the boiler, and cause the formation of acids, whose action, if not checked, induces a great amount of corrosion and pitting. As pitting, if once started, is exceedingly hard to stop, the boiler water should be frequently tested for acidity by means of litmus paper.

If such boiler-feed waters are discovered to be acid by this test, soda or soda and lime should be added to the feed water, preferably in small quantities administered every day, although in engines which under regular conditions will run almost without internal lubrication the use of soda need not be so

frequent. In one very bad instance of pitting and corrosion, a set of boilers which was attacked in this way was always opened out when in port and the corroded surfaces were scraped clean and washed with soda after being coated with lime-wash. This was done regularly, the result being that the pitting did not get any worse. In order to prevent oil from entering the boiler, even if it is necessary to use it in the engine lubrication, it is advisable to fit efficient oil filters in the feed-water service, so as to intercept any particles of oil before passing to the boiler. For seagoing practice, filters employing cores are the most suitable, as the filtering medium can be readily changed, even on the longest voyage.

One of the most fruitful causes of corrosion and pitting is galvanic action, and this is prevented by the use of zinc plates hanging in the water space of the boiler. These plates are attached to suitable hangers, having a proper metallic connection with the plates of the boiler. They should also be attached to the stays, furnace tubes and combustion chamber stays. Galvanic corrosion is thus transferred to the zinc, which is thus eaten away and the iron and steel preserved. It is found in practice that zinc plates are better than zinc balls connected to the boiler parts by wires, which is a frequent arrangement, inasmuch as the wires do not make a perfect connection.

It may be mentioned that although there are at the present time many liquids manufactured for the purpose of preventing boiler corrosion, these are not generally used in marine practice, and, generally speaking, it is hardly advisable to adopt them except under very careful skilled advice. Some of these liquids, as a matter of fact, do considerably more harm than good.

External corrosion of the boiler is a matter which can to a greater extent be more easily and immediately detected than internal corrosion. It can, as a rule, be prevented by keeping the mounting glands tight, and by seeing that the surface is thoroughly treated with a good composition or a covering of red lead and paraffin. The mattresses or boiler coverings

should always be kept in good order and condition, and if attention is paid to these points of up-keep it is probable that the danger from corrosion will be very largely minimized, if not altogether obviated.—"International Marine Engineering."

ELIMINATION OF OIL FROM CONDENSED STEAM.

The recent article in "Power" by Geo. H. Gibson relating to the elimination of oil from condensed steam has stated the facts of this important subject in a very clear and interesting manner, giving the true cause of the difficulty heretofore found in thoroughly and completely removing the last particles of oil from the cloudy and emulsified water. This difficulty, as Mr. Gibson stated, has been owing to the extreme minuteness of the oil globules found in the emulsified or cloudy-looking condensate, and which may be as small as one-one-hundred-thousandth inch in diameter, defying all attempts to remove them by any heretofore known filtering medium such as turkish toweling, terry cloth, sand or other permeable substance. It will therefore be interesting to readers, and steam users in general, to know that a newly discovered process has lately been put into practical operation, which is based upon the energetic absorptive power or physical attraction which some natural magnesium silicates have for oil or grease, as discovered by the writer some years ago.

These absorbents consist mainly of the pulverized rock from serpentine quarries, found in various parts of the United States and Canada, which is mixed with a certain proportion of sand to increase the surface contact with the condensation water while the latter is passing through a bed of the mixture, so that the smallest emulsified oil particles are immediately absorbed and held back, rendering the water clear and free from the last traces of oil or grease, and as bright as any sparkling spring water or distilled water. The amount of oil which this fibrous serpentine will absorb is very large, often more than 50 per cent. of its weight, and the tenacity with which the oil will be

retained within the absorbent is such that it can be removed only by means of benzine, gasolene or other similar solvent, boiling water or steam having very little effect thereon. The apparatus used in working this process is almost the same as that of an ordinary pressure filter, except for the special connections necessary for stirring up the absorbent mixture from time to time. There is no skilled labor or expert attendance required and no danger in using an excess of the mixture such as there would be in chemical coagulation. The cost of the material is very reasonable, consisting mainly in freight charges to the various localities where required. The thickness of the bed of absorbent material may be from 6 inches to 6 feet or more if necessary, depending upon the pressure available, and the filters may be adapted for either land or marine use.

As an evidence that the purifying action of the fibrous serpentine is not a filtering one it may be stated that when pulverized, one part of the substance in 2,000 parts of average emulsified water will clarify the latter by simply shaking or violently stirring, or boiling up the contents of the vessel. After standing half a minute the water will be found entirely clear and free from oil. At the works of Alex. Miller & Brothers, Jersey City, where it has been in successful operation over six months, this process may be seen and inspected at any time. Wherever the above-described absorbent serpentine process is used it is, of course, also desirable to first remove the free or coarse oil by means of skimming tanks or ordinary terry-cloth filters so as to utilize the absorbent serpentine in the most economical manner, leaving the latter to remove the milky or cloudy appearance of the condensate which the terry cloth or Turkish toweling is incapable of filtering. The separator in the exhaust mains should also be retained in any case as stated by Mr. Gibson, but not because they could ever be depended on to remove sufficient oil from the steam to make the latter safe as a feed water after condensation (which they can never do) but to keep as much oil as possible out of the condenser and the hot well so that the

subsequent removal of all of the oil from the main condensed water will be made simpler.

A great drawback about chemical coagulation is the expert attendance necessary to prevent too much or too little of the reagents being introduced into the water. Unless the proper and exact proportions are used to meet the varying quantities of emulsified oil in the condensation the resulting water will retain too much soluble and detrimental matter in solution or there will not be sufficient precipitation to remove all of the emulsified oil. The absorbent serpentine process here described, being of a non-chemical nature, an excess of clarifying material used can have no detrimental effect as the material is practically insoluble. All expert chemical attendance is therefore unnecessary, and the cost of working consequently nominal. By this absorbent process, furthermore, all the drip can be saved as well and readily as the regular condensation, so that there is no waste of water whatever except the small amount lost by leakage or evaporation. Careful and frequent analyses of condensation water clarified by the herein described absorbent-filtration process have shown a total absence of oil or grease and a mineral residue in solution of but two to three parts per 100,000, consisting mainly of magnesium carbonate, a light harmless substance which has no tendency to form scale and only about sufficient in amount to neutralize any fatty acids that may have been produced by the partial decomposition of some of the lubricating oil in the condensed water.

In the removal of oil from condensation or from steam the essential requirement is to remove *all* of the oil, not *nearly* all or not even 99.9 per cent., as is frequently stated in trade publications. Unless the remaining one-tenth of one per cent. is also removed there will still be oil enough left to leave the water dangerous to use for boiler feed. This fraction of a per cent. of oil remaining in the steam is plainly visible in the subsequent condensation, giving the latter that milky, opalescent appearance mentioned in Mr. Gibson's article, which is always indicative of oil or grease, no matter how often it may have been filtered through terry cloth, excelsior, Turkish tow-

eling or any other simple filtering material, and the water if used in this condition will be certain sooner or later to show the usual evil results in boilers, particularly in those of the water-tube type.

Concerning the influence of even small amounts of grease such as may be found in milky-looking or opalescent condensation, the following may be instructive and serve as a warning against the use of such water for boiler feed. Messrs. Strohmeyer and Barron, in a paper before the Institute of Naval Architects in Manchester, on "Distortion in Boilers Due to Overheating," state that the peculiarity of grease deposits in boilers is that their effect is out of all proportion to their thickness.

Scale one-eighth of an inch in thickness will raise the temperature of furnace plates 300 degrees F. As grease offers ten times more resistance to heat, one would think that one-eightieth of an inch would have the same effect as this thickness of scale, but experience shows that the merest trace of grease certainly less than one-thousandth of an inch, or one-tenth of the above, can cause far more serious injury than scale. It also follows from the above statement that a boiler containing the slightest film of grease on its inner surfaces must have a greatly diminished efficiency as a steam producer. In any case it is highly desirable to remove every trace of grease from the feed water. Where scale also is present the effects of the grease are intensified.

As above stated, this emulsified oil could not be removed by any of the filters heretofore on the market; the most that the latter could do was to remove the coarse or free oil which will readily rise to the surface of the water and can then be skimmed off. Your correspondent, John S. Kirk, in the October number of "Power" also emphasizes this clearly by stating that "no amount of filtering which is practicable in all ordinary steam-power plants will separate this emulsion into its elements of water and oil. They go to the boiler together, and while the water is made into steam the oil is left behind to stick to the tubes and shells and make all kinds of trouble."

It is also a well-known fact that grease separators in the exhaust-steam pipe, although efficient for removing a very large proportion, particularly when new, do not remove the last traces of grease.

In the opinion of a great many practical men who have from time to time examined the absorbent serpentine process for removing the last traces of oil from condensation feed water as herein described, this problem is finally and practically solved, and should be of great and particular interest to those who now hesitate about installing surface condensers, owing to the impossibility heretofore of totally removing the oil from the condensation when used for boiler feed, as well as to those who have had costly experience resulting from oily feed water.—ARTHUR E. KRAUSE, in "Power and The Engineer."

THE FUTURE DEVELOPMENT OF THE STEAM TURBINE.

We had occasion last week, when reviewing the work of the year, to refer to the steady increase in the number and power of Parsons steam turbines adopted in the merchant marine and Royal Navy, and also to the improved economy, from the point of view of steam consumption, of recent marine installations, as established by our later warships on trial. An equally important improvement is being achieved in turbines constructed for driving electric generators in central stations; and as these data indicate the lines along which future developments will be made, no excuse need be made for directing further attention to this subject.

When the trials of the famous Elberfeld turbines were made in 1900, the steam consumption, which in the best test amounted to 19.42 pounds of steam per kilowatt-hour, constituted a record, and was, indeed, a very remarkable result in view of what would now be considered the moderate dimensions of the unit and the fact that it was the first of the size to be constructed by its builders. Station engineers nowadays have, however, less generous views as to where the dividing

line should be drawn between a merely healthy and an abnormal appetite for steam. Improvements in condensers have largely helped the turbine builder to meet the increasing demands of his customers in the matter of steam economy. In the early days the importance of a good vacuum to turbine efficiency was little realized by station engineers, and builders used to pray for a low barometer during the period of the reception tests, as few ever bothered to reduce to absolute pressure the readings of the vacuum gauge, so that the turbine obtained its rated consumption with a vacuum which, to all appearance, was nothing out of the common. The growing use of superheat has also contributed not a little to the success with which higher steam economies have been obtained, but probably more important than either has been the increase in the size of the units.

As a result of these different factors, for instance, Messrs. Howden & Co., with the 6,000-kilowatt Zoelly turbine, at Manchester, have been able to generate 1 kilowatt-hour with the equivalent of 13.8 pounds of steam, supplied at 186 pounds pressure, 140 degrees superheat, and exhausted at a 28-inch vacuum. Even better results in actual steam consumption have been recorded with the 9,000-kilowatt Curtis set at Chicago, where the actual steam consumption was under 13 pounds per kilowatt-hour. Figures of this kind require, however, adjustment for the various factors involved, and the actual efficiency ratio (governor-valve to switchboard) at Chicago was about 62 per cent., the remarkably low consumption of steam being due to an equally remarkable vacuum of 29½ inches, corresponding to a condenser temperature of only 60 degrees. F. The efficiency ratio of the Zoelly turbine at Manchester is stated to be 69.2 per cent., reckoned from governor valve to switchboard; and in a paper read before the Association of Engineers-in-Charge on December 1, Mr. G. B. Williamson has claimed this figure as a record. It would, however, appear to be surpassed by the figures obtained with the new 6,000-kilowatt units installed at Lot's-road by Messrs. C. A. Parsons & Co., where, with very similar conditions at the stop valve

and a vacuum of 27.7 inches, the consumption was 13.7 pounds per kilowatt-hour.

Theory indicates that, leakage losses apart, the reaction turbine should have a sensible superiority, and the close results of the rival systems seem to indicate that the greater efficiency of the reaction turbine at the low-pressure end is discounted by the losses by leakage in the high-pressure section, where the steam is dense and the blades are short. A combination of an impulse high-pressure end with a Parsons low-pressure section may, therefore, ultimately prove to be superior to either type pure and simple.

The Melms-Pfenninger turbine is, of course, constructed on these lines, and has given very good results. There are, however, some practical advantages in the adoption of a velocity-compounded wheel for the impulse section, which may, in certain cases, lead to this construction having the preference in spite of its lower efficiency. Where very high pressures and great superheats are to be used, the use of a velocity-compounded impulse wheel makes it possible to confine the extremes of pressure and temperature to a mere nozzle box, in place of admitting them to the shell of the turbine proper. Large turbines, very much on the lines suggested, are now being constructed by the Allgemeine Electricitats-Gesellschaft, in which a single velocity-compound wheel constitutes the first stage of the turbine, whilst the low-pressure end is a drum fitted, however, with impulse blades, though there is much reason to believe that the Parsons system would here have considerable advantages, its leakage losses being no greater, whilst its blade efficiency is higher. The advantages of the particular combination suggested belong rather to the facility it gives for the use of extra high-pressure steam than to the attainment of exceptional efficiency ratios. If, however, a turbine is constructed so that it can utilize a heat drop of 450 B.T.U. with an efficiency ratio of 64 per cent., it will exceed in steam economy another turbine having an efficiency ratio of, say, 70 per cent., but capable of utilizing down to the same condenser pressure but 380 units of available heat.

This distinction between steam economy and efficiency ratio is well illustrated by recent marine practice. The steam consumption of the most recent Curtis marine turbine as fitted to a battleship is very little more at full power than that recorded with the *Dreadnought*. The latter, however, ran with its steam wire-drawn in the steam chest down to 150 pounds, though the boilers were supplying the steam at 240 pounds; whilst in Curtis sets, at full load, practically the full boiler pressure is generally admitted to the turbine. With not much difference in actual fuel consumption, therefore, the efficiency ratio of the reaction turbines must have been considerably the greater. This drawback, which is inevitably associated with the principle of velocity compounding, is largely discounted by the facility with which large quantities of "available heat" can be dealt with. By the adoption of blade speeds exceeding 600 feet per second the Allgemeine Electricitats Gesellschaft have thus secured excellent steam economy with a turbine having two pressure stages only. Blade speeds have also been raised in impulse turbines having no velocity stages, and in units of large size are now often about 400 feet per second. High-blade speeds involve, of course, shorter and cheaper turbines. With the reaction type, the advance in this regard has been less marked, but the question as to whether it will be prudent to reduce the factors of safety against centrifugal forces seems worthy of consideration. So far as the blades themselves are concerned, the safety, even at greatly increased speeds, seems absolute if they are caulked in the usual way; though some of the systems which involve the use of a foundation ring are perhaps less satisfactory in this regard. With these, the drum is cut away more, and the "dead" weight to be carried by it is increased. A properly-caulked blade will break in two before it will pull out. As to the drum stresses, again, these could probably be raised without danger of a burst. The load on the drum is perfectly steady; and if the emergency governor fails to act in a runaway, a complete smash-up is inevitable, whatever the normal working stress may be. The real difficulty of using higher surface speeds

would seem to be due rather to questions of distortion than of strength, and such increase might necessitate larger clearances in the dummies and over the blade tips. Cases have been known of the unsupported ends of marine turbine dummies bellmouthing under the centrifugal forces. Of course, if the drum be replaced by solid disks strung together on the shaft, each carrying two or four rows of blades only, the centrifugal forces can be met as fully as desired, but such an arrangement sacrifices the straightforward mechanical simplicity of the Parsons type of machine.

The small clearances often raised as an objection to the Parsons turbine appear to have lost their terrors with the introduction of the blades having thinned tips, and the system of assembling the blades in sections, which are then transferred bodily to the turbine body, has made it possible to silver solder a lacing wire to even the shortest blades. This was previously impossible, since, in the case of short blades, the heat was conducted away too rapidly into the heavy mass of metal constituting the drum or the casing.

Had these or equivalent improvements not been effected, there would have been little occasion for surprise if the reaction type had had to give way to the impulse pattern. Some of the latter have, no doubt, had their troubles, but these were relatively less serious than with some of the earlier patterns of the Parsons machine built before designers had realized the difficulties which supervened on the general introduction of superheated steam. Had these not been surmounted, the lower cost of construction and the higher efficiency ratio possible with the reaction type would hardly have saved it from supersession. It has undoubtedly been more difficult to construct free from risks of injury than was the impulse type, and variations made from the original designs by licensees and others have often been injudicious. As matters stand, however, the best designs of both types are now probably equally reliable—certainly the reaction type stands at no disadvantage—and their relative importance in the future will be settled mainly on considerations of price. Cases, no doubt, are still recorded

of reaction turbines giving trouble, but these are invariably old patterns, in which sufficient provision was not made for the distortions caused by superheat. Bronze foundation rings, for instance, if used at the high-pressure end of a turbine, are liable to work out if the superheat is considerable, owing to differential expansion. By substituting steel rings here the difficulty disappears.

In marine work it is necessary to distinguish between the case of the mercantile marine, where boats are driven at constant speed, and the Navy, where most of the steaming is done at a speed very much below the maximum. So far as data are at present available, it would appear that for the former condition of affairs the reaction type weighs less for a given steam economy at full load than its rival, though the matter is complicated by the fact that the comparison has to be made between two-shaft and three-shaft or four-shaft systems. In naval work the impulse type seems to provide for cruising speeds more simply than has yet been done with the reaction type. This may be altered by the introduction for this service of Mr. Parsons' partial-admission reaction turbine; but, failing that, we should not be surprised to see the adoption of a combination system consisting of a number of high-pressure velocity stages followed by a low-pressure reaction turbine.—"Engineering," Jan. 7, 1910.

THE SCHULZ STEAM TURBINE.

Notwithstanding the fact that the Schulz steam turbine has been in existence for some years, it is probably not so well known in this country as it is on the Continent, and in view of the fact that it has been steadily gaining prominence abroad, some particulars of its general construction and the special advantages claimed for it may prove of interest. Recognizing that multi-stage turbines can only work economically with a full supply of steam, and that a diminution of the admission of the steam into the first stage occasioned by a reduction of the

load is accompanied by a comparatively poor efficiency of the work of the steam in the turbine, Schulz set himself the task of designing a turbine which would make the most of the expansion of the steam for small loads as well as for large ones. A turbine of this type, which was used for propelling a small boat in 1900, is shown in the drawing Fig. 1. It is a compound multiple-expansion impulse parallel-flow turbine. The high-pressure section is seen on the right and the low-pressure and reversing turbines are on the left. Steam passes alternately through guide rings and through moving blades. Slides, which

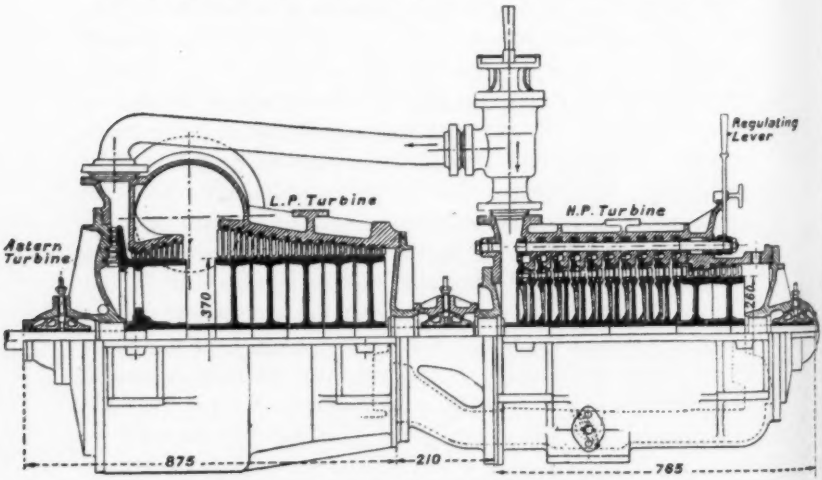

Fig. 1.—SCHULZ MARINE STEAM TURBINE.

will be dealt with more fully later, are provided, which act to limit the passage in the guide rings so as to control the flow of steam through the casing, according to the load. It will be noticed that the exhaust of both the high-pressure and astern turbines finds its exits through the same channel which is now the customary construction in other turbines. A drawing of the latest type of Schulz turbine suitable for marine propulsion is shown in Fig. 8, where the astern turbine is also provided with regulating gear, and there is no separate low-pres-

sure casing. Before entering upon the details of construction, attention may be called to a test which was made on a 650-horsepower Schulz turbine, built by A. G. Wesser, of Bremen, the results of which are shown in the curves Fig. 9. The upper curve in this gives the peripheral speed of the rotor, the middle set of curves the steam consumption, and the lowest curve the

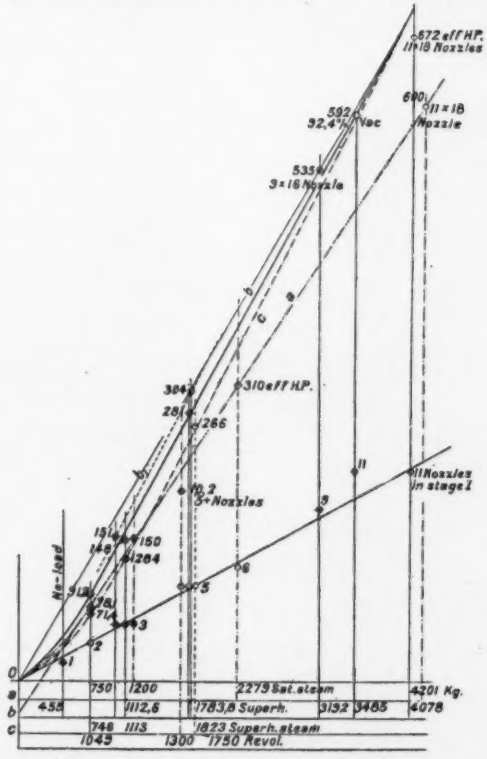

Fig. 2.—Curves of Nozzles and Speeds.

revolutions per minute. From these curves it will be seen that the steam consumption decreases with an increase in peripheral speed. If the performed work falls below 6/9 of the full load, the steam consumption per horsepower increases. The curves given in Fig. 2 show the number of nozzles used for the differ-

ent speeds and the resulting horsepowers. The curves b and b_1 and c were plotted from results obtained with superheated steam; but in the case of curve c the speed was less than in the other instances. The degree of superheat was 100 degrees C. The curve a was plotted from results obtained with saturated steam. It will be gathered that the efficiency of the Schulz turbine at light loads is very good. In order to make the most of the expansion of the steam for small loads as well as for large loads, adjustable slides are arranged in the various stages in front of the inlet openings of the guide-vane wreaths. As will be gathered from Fig. 10, the regulating device consists of circular segments operated by spur wheels, which are mounted on a spindle on top of the turbine, and in this way the fall of pressure available is fully utilized, even at the smallest loads. The regulation, according to this method, is carried out in such a manner that the steam does not work within the same fall of pressure at the different loads. For the smaller loads more of the guide passages are opened in the later regulated stages in proportion to the number opened in the first stage, so that the section of the open guide passages increases at a greater rate for small loads than for large ones. A 560-horsepower turbine, working with a steam pressure of 13 atmospheres and 90 per cent. vacuum, was tested with different loads, and the results obtained were as follows:

Load	$\frac{1}{1}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{6}$
Drop in load, per cent	..	25.0	50.0	75.0	83.3
Kilos. of steam used	6	6.15	6.35	7.7	8.25
Increase in amount steam used, per cent.	..	2.5	5.5	28.0	37.0

As already stated, and as shown in Fig. 8, in the latest type of Schulz marine turbine regulation is also carried out in the astern turbine, so that the steam consumption in this turbine may also be a minimum. The regulation for the ahead and astern turbines occurs simultaneously, so that the astern turbine is prepared in advance for its work in the same way as the ahead turbine, which is a great help to the engineer.

The rotors of these turbines are of forged steel, and the vari-

ous sections fit on to coned portions of the shaft, and they are secured by means of keys. The blades are not fastened directly into the stator, but into a special ring, which fits inside, as shown in Fig. 3. In the construction *a* the nozzle is com-

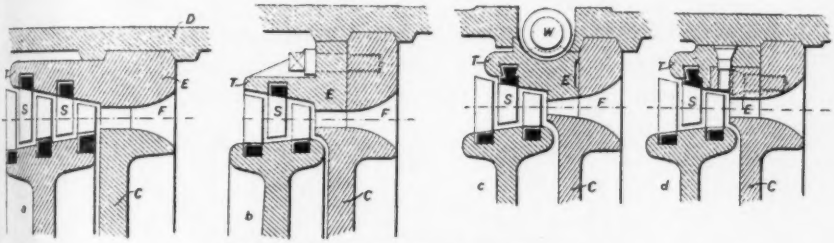

Fig. 3.—ARRANGEMENTS OF NOZZLES.

posed of a ring mounted on the partition C. The ring E completes the nozzle F of the disc, which latter can be revolved on the shaft after removing the upper half of the turbine casing D. The carrier or support T for the blades S is mounted on this ring E, and in this case it is shown in one piece with the ring. If the nozzles F are only distributed over a portion of the periphery, radially divided annular pieces, each of which is provided with a guide-blade carrier T, are connected with the partition C. In the construction *b* the ring E is mounted by means of screws on the nozzle ring on the partition C. In the example *c* the annular piece E, which forms a portion of the nozzle situated on the partition C, is adapted to be displaced relatively to the ring by means of a worm W engaging in teeth on the annular piece. The annular piece in this case serves as a regulating valve for altering the expansion of the steam, for the blades in connection with the exit part of the nozzles are displaced with the latter. Lastly, in example *d* the support T of the blades S is attached by means of screws to a separate ring or annular piece E, which latter forms a portion of the nozzles in the ring C.

The advantages claimed for these constructions are that the work on the interior of the casing is simplified. The casing retains its full strength, because large apertures are unnecessary,

in order to allow of the insertion of separate blade supports. After the removal of the upper portion of the casing D the blade supports mounted on the disk may be turned on the turbine shaft without the latter, together with the blade wheels and disks, having to be removed from the casing. After the lower halves of the disks have been turned upwards the parts fixed thereto may be easily inspected, removed, and repaired. The blades are made of Delta metal, and as shown in Fig. 4

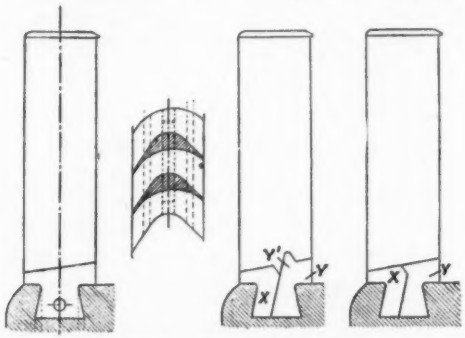

Fig. 4.—TURBINE BLADES.

they are inserted in dovetail-shaped annular grooves of the wall of the carrier and of the running wheel, intermediate caulking pieces being inserted between the blades. The feet of the blades and the intermediate caulking pieces are provided with perforations, and after insertion in the grooves a mandrel is driven into the perforations of several of the blade feet and intermediate caulking pieces, simultaneously pressing the whole closely together. Increased safety is obtained by driving in pins after the removal of the mandrel. As it is not possible to insert a mandrel in the case of the last few blades, wedges are used as shown on the right of Fig. 4. The parts Y are first inserted, and then the parts X are driven in as a locking piece and then secured by hammering down the projections Y^1 of the parts Y on to and over the parts X, so that while the parts X lock the parts Y in the first place, the parts X are in turn locked by the projection Y^1. According to this method of at-

taching the blades in annular grooves, the feet of the blades need not be closely fitted beforehand to the undercuttings of the grooves. They may, therefore, be easily inserted, and the pins cause the feet of the blades to expand and securely bind themselves. Steam nozzles are designed for producing a continuously broadening steam jet which acts advantageously on the rotors. To ensure against leakage the stuffing boxes of the main shaft are packed, as shown in Fig. 5. They are provided

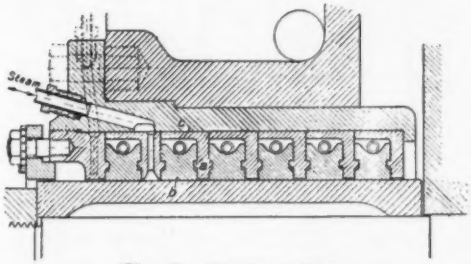

Fig. 5.—STUFFING BOX.

with a number of rings which are held upon the shaft by means of springs. The rings *b* are held against the stationary intermediate rings *c* by another series of rings *a*, so that the rings *b* are just touching the surface of the shaft. In this way the binding effect of the rings is avoided and the wear minimized. The intermediate bronze rings *c* take up the weight of the packing, but in spite of this the latter could take care of any unusual pressure upon the shaft. The packing rings cannot turn with the shaft, as the springs which hold them against the shaft also prevent them from turning.

The regulating gear already referred to, and shown in Fig. 10, consists of segments of rings *a*, in which the movable plates *b* are mounted. These plates are held against the nozzle openings, and the plates are held in place by projecting ledges, as shown in Fig. 6. These controlling rings or valves may be actuated by a spur wheel arranged as shown in Fig. 10 or as in Fig. 6. A scale is provided to indicate the position of the slides. In order to press the packing ring *b* on the seats,

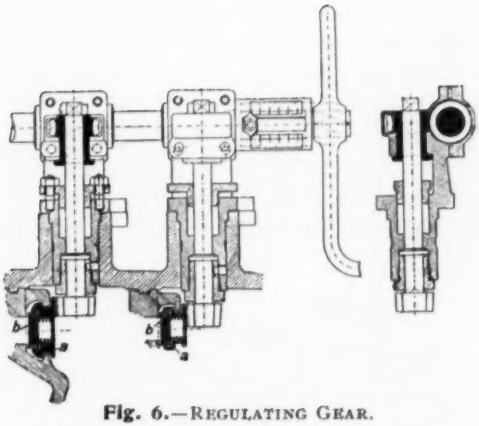

Fig. 6.—Regulating Gear.

springs are employed, as indicated in Fig. 6. As is customary with other marine installations, the power may be distributed over two or four propeller shafts. Fig. 7 shows the arrangement of a plant for a battleship of 50,000 horsepower, the maximum speed being 25 knots. The speed of the propellers

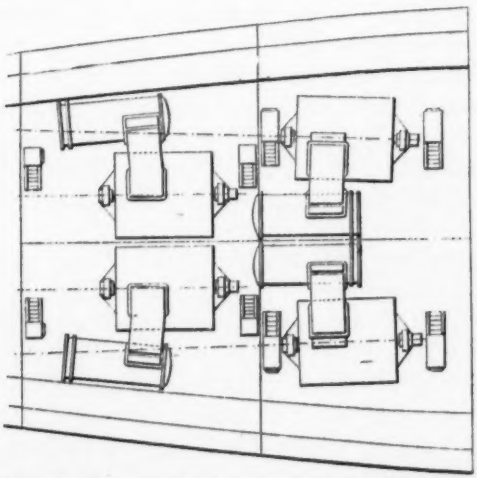

Fig. 7.—Turbine Arrangement for Battleship.

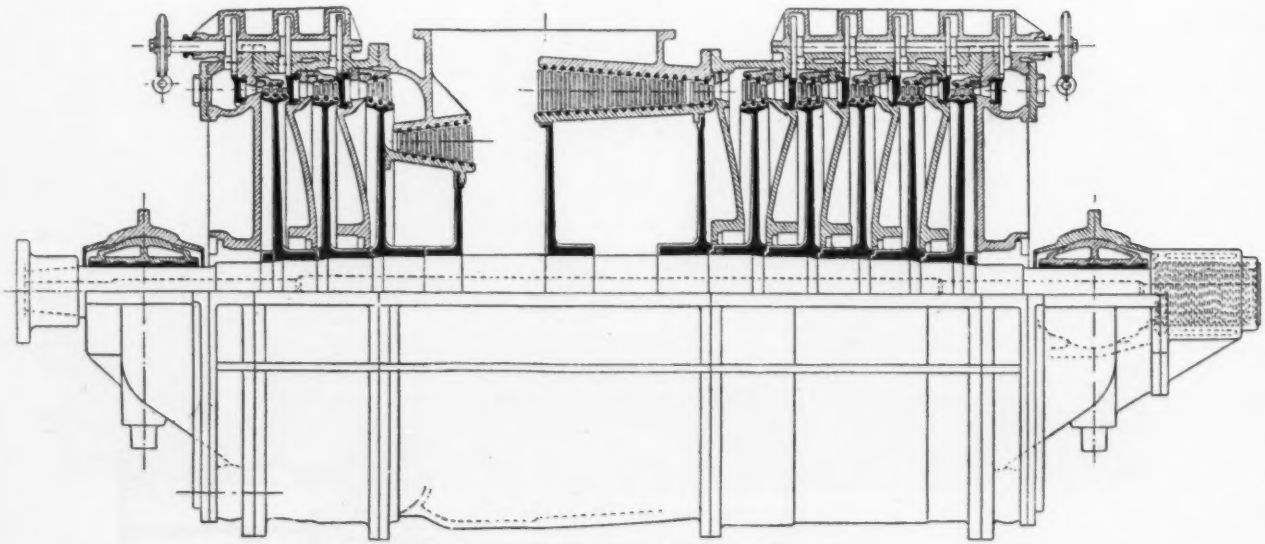

Fig. 8.—Latest Type of Marine Turbine.

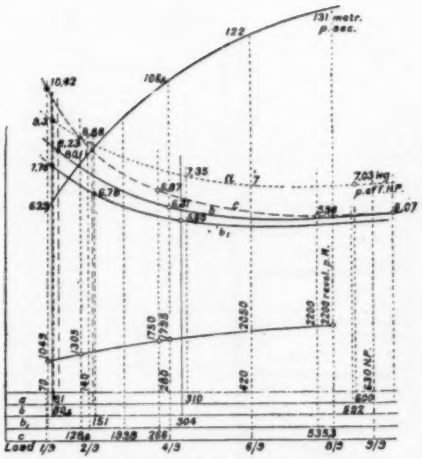

Fig. 9.—RESULTS OF TESTS.

is 230 revolutions per minute. Another arrangement where the power is distributed over four shafts is shown in Fig. 13. The arrangement is practically self-explanatory. It will be seen that two of the astern turbines are built in separate casings, whilst the other two are built within the low-pressure turbine. This is a 22,000 horsepower installation designed to distribute the power over four shafts.

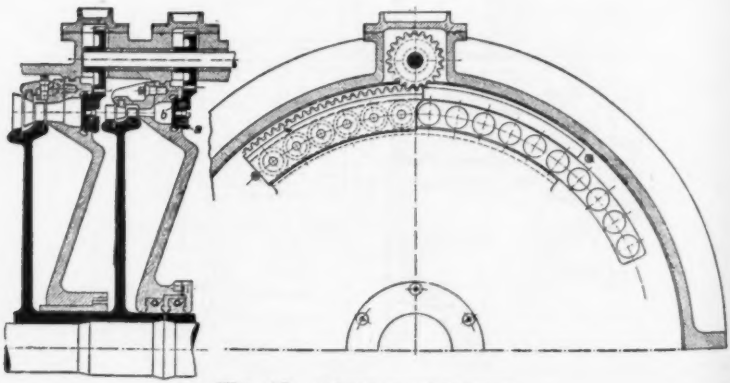

Fig. 10.—REGULATING DEVICE.

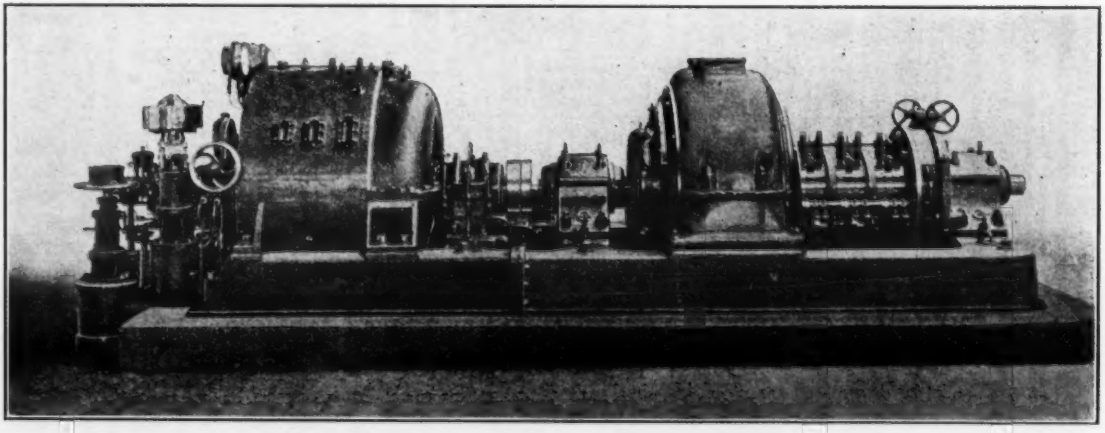

Fig. 12.—110-H. P. Steam Turbine and Dynamo.

A twin-screw arrangement for a 22,000 horsepower vessel is shown in Fig. 11. The turbines are similar to that shown in Fig. 8, with the astern turbines built in the same casing; there are, however, more stages than in the turbine shown in Fig. 8. This arrangement occupies a minimum amount of space. In both arrangements — Figs. 11 and 13 — the

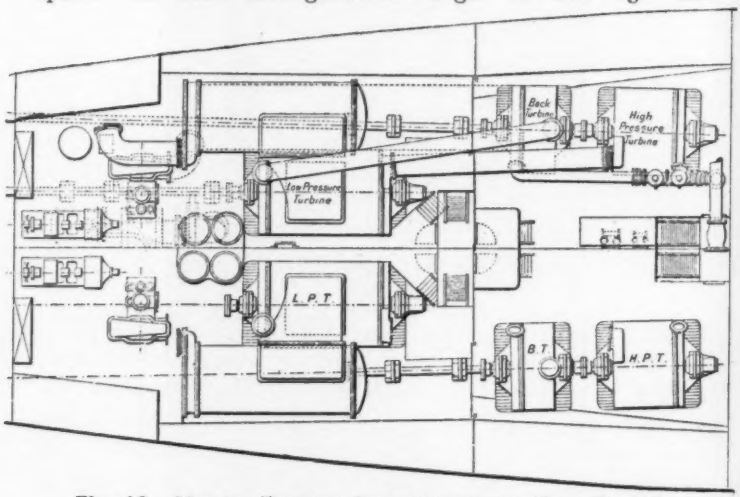

Fig. 13.—MARINE TURBINE ARRANGEMENT ON FOUR SHAFTS.

amount of space available for the plant is the same. The management of both arrangements is said to be very simple, the astern turbines responding remarkably well to a change from full speed ahead to full speed astern. Detailed specifications for these two installations are given below:

	Fig. 13.	Fig. 11.
Number of propellers	4	2
Diameter of propellers, mm	2,380.0	3,370.0
Total area of blades ÷ disks	1.17	1.175
Revolutions per minute	430.0	313.0
Peripheral velocity per second, m	54.0	55.0
Maximum indicated horsepower	22,000.0	22,000.0
Slip, per cent	22.0	22.0
Indicated thrust, kg	99,800.0	99,800.0
per 9 cm. blade area, kg	1.035	1.03
Mean diameter of turbine rotors, mm	2,000.0	2,700.0
Peripheral velocity of rotors per second, m	45.0	44.2

The general construction of the ahead and astern turbines is the same whether the steam passes first into the high-pressure turbine and then into the low-pressure turbine, or whether it simply passes into one turbine and expands from the initial pressure to vacuum. The steam first acts upon a number of partially-bladed disks, and then upon others which are fully bladed. The astern turbines are generally able to develop 80 per cent. of the power developed by the ahead turbines. The steam consumption at different speeds for the two above-mentioned installations is given below:

Speed, knots	17.0	25.0	26.5
Steam pressure		15 to 16 atmospheres.	
Superheat, degrees C	40.0	0.0	0.0
Back pressure		0.06 to 0.09 kg.	
Effective horsepower	4,200.0	18,000.0	22,000.0
Steam consumption per effective horsepower hour, kg	7.1	6.0	6.1
Steam per hour, kg	29,820.0	108,000.0	134,200.0
for auxiliaries, kg	4,300.0	10,300.0	10,800.0
Total steam, kg	34,120.0	118,300.0	145,000.0
Coal (9-fold evaporation), kg	3,800.0	13,540 0	16,200.0
per effective horsepower, kg.	0.9	0.752	0.736

Of the two arrangements shown in Figs. 13 and 11, the latter is considered by far the best, as there are only two separate turbines, as compared with six in the former arrangement.

The Schulz turbine as constructed for dynamo driving naturally differs somewhat to the marine turbine owning to the higher speeds at which turbo-driven electric generators can be run. It consists of an impulse turbine having several stages, all of which are in one casing, as is generally the case with other types. An illustration of a small set of about 110 horsepower, and running at a speed of 3,400 revolutions per minute, is shown in Fig. 12. Turbines of this class have been built for 2,200 revolutions per minute, in which case there are four stages per 500 horsepower; also for 2,000 revolutions per minute with five stages per 500 horsepower. In the latter case the first three stages may be regulated. These turbines are designed to generate 250 kilowatts. Larger turbines of 1,400

horsepower run at 1,500 revolutions per minute, and three stages are regulated, the total number of stages being five. Automatic emergency governors are fitted for shutting off the steam in the event of the speed exceeding a predetermined value.—"The Engineer."

INTERNAL-COMBUSTION ENGINES FOR MARINE PROPULSION.*

The principal advantages claimed for the internal-combustion engine are economy in fuel, weight and space. It is generally conceded that the least economical of internal-combustion engines is more economical than the best steam engine, and there are abundant data to substantiate this claim. This factor alone would seem to make it more than probable that the internal-combustion engine may play an important part in the future of marine engineering.

On the question of economy of weight and space very little data are available. What little there is is only applicable to small units, and it by no means follows that the results obtained in small units can be applied directly to larger installations. It is to a general consideration of some factors affecting weight and space that the present paper is directed.

Some very remarkable results have been obtained with small petrol engines in respect to power developed on a given weight, but it is to be remembered that this extreme lightness is due principally to two causes, viz: (1) a high speed of revolution, and (2) the use of special materials of construction. No part of the extreme lightness is due directly to the engine being of the explosive type. The word "directly" is used advisedly, because, as will be seen presently, there are certain conditions incidental to the internal-combustion engine which make higher speeds of revolution possible than are possible or advisable with a reciprocating steam engine of similar size.

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*Some considerations on the application of Internal-Combustion Engines for Marine Propulsion. Paper read before the Institution of Naval Architects, April 1, 1909, by H. C ANSTEY.

To arrive at a proper sense of proportion on the question of weight and space, it is desirable to examine it on broad mechanical principles. Of the factors which together make up the horsepower of an engine, the only one which in an engine of given power may be altered without affecting its weight is the number of revolutions, subject always to the consideration that the inertia forces due to the speed are not such as to require special strength and weight of parts and foundations.

If we suppose the mean pressure for a particular engine to be constant, and the product of the remaining three factors—viz: area of cylinder, stroke and revolutions—constant also, it will follow that any increase in speed of revolution will be accompanied by a reduction in area of cylinder, or in length of stroke, or both, and owing to these reductions a saving in weight will be obtained. For an engine of given power, therefore, increase in speed of revolution is accompanied by a reduction in weight per horsepower, or, put conversely, an increase in the horsepower per ton of engine weight. Some years ago the author, in examining a number of machinery weights of His Majesty's ships, found that the engine weight could be divided into two parts—one proportional to the horsepower, and the other proportional to the horsepower divided by the number of revolutions. Taking a mean of weights for all the vessels of the same class, in order to eliminate differences in design, the engine weight could be expressed as—

$$\frac{\text{I.H.P.}}{k} + k_1 \frac{\text{I.H.P.}}{N},$$

N being the number of revolutions per minute.

The values obtained for the constants k and k_1 gave a close approximation to the actual weights in several classes of vessels, and it was possible to determine with some degree of certainty what increase in weight would be involved in a new design by lengthening the stroke and decreasing the number of revolutions. As to the first term, it is clear that in a steam engine there are items of weight, such as steam pipes, valves,

condensers, and many others, which will depend upon the weight of steam passing, and hence directly upon the horsepower. The existence of the second term is accounted for by the reasoning previously given.

As these elementary principles are purely mechanical, and independent of the fluid employed, the formula given above, with suitable values for the constants depending upon the type of engine, will no doubt apply to any reciprocating engine.

Returning to the factors in the formula for horsepower, if we assume piston area, stroke and number of revolutions to be constant, the power is proportional to the mean pressure. The weight of the engine is, however, proportional to the maximum pressure for which the engine has to be designed, and the horsepower per ton is, therefore, proportional to the ratio, mean to maximum. In considering what this ratio is in a marine steam engine we are faced with the difficulty that the power is divided between several cylinders, each having a different ratio of maximum to mean, but for purposes of comparison we shall probably not be far wrong if we assume that the whole of the expansion is carried out in the low-pressure cylinder, which develops the whole power, and is credited with the whole of the weight. Under this assumption the maximum pressure will be, say, 250 pounds, and the mean 50 pounds, giving a ratio of maximum to mean of 5. This ratio will vary with the ratio of expansion, and will be generally higher in the merchant service than in naval practice. In internal-combustion engines, in spite of variety of type, there is not very great variation in this ratio. In the engine with which the author is best acquainted it is approximately 4. In petrol engines it will be rather less, and in engines using a high compression it will be somewhat more. Taking the ratio as 4, it must be corrected for the cycle employed, and as the internal-combustion engine has in most types only one working stroke in four, the ratio must be multiplied by four, and the comparative figures will then be: for the steam engine, 5; for the internal-combustion engine, 16; that is to say, considered from the point of view of pressure alone, the horsepower per ton in an internal-combus-

tion engine will be 1/3.2 times that of a steam engine of the same linear dimensions.

There are, however, certain factors to be considered which will make the comparison more favorable to the internal-combustion engine. First, the items of weight in a steam engine which are directly proportional to the horsepower account for a fair proportion of the total weight, but in the internal-combustion engine the proportion of similar parts will be very much less.

Steam pipes and condenser, for example, have a counterpart in the air and exhaust silencers, which are not pressure parts, and may be made a quite inconsiderable part of the total weight. Inlet and exhaust valves also will be lighter in proportion than the slide valves of a steam engine. Secondly, the single-acting engine of the trunk type is lighter than a double-acting engine of the same diameter and stroke, as the latter requires additional height and heavier parts, due to the piston rod and crosshead; hence the horsepower per ton will be, in a double-acting engine, not quite four times what it will be in an engine of the same linear dimensions working on the four-stroke cycle. Thirdly, the internal-combustion engine is capable of a higher continuous speed than a steam engine, for two reasons: (1) lighter parts, and (2) lower mean pressures on the bearings. With regard to (1), the following are some weights of reciprocating parts expressed in pounds per square inch of piston area:

High-pressure steam cylinder,	8	inches diameter, .	2.95
High-pressure steam cylinder,	33	inches diameter, .	5.46
High-pressure steam cylinder,	40	inches diameter, .	7.7
Petrol engine, ...	$4\frac{5}{16}$	inches diameter, .	0.63
Oil engine, ...	$9\frac{3}{4}$	inches diameter, .	1.45
Oil engine, ...	15	inches diameter, .	2.2
Gas engine, ...	24	inches diameter, .	4.0

With regard to (2), the sizes of journals in internal-combustion engines are larger than those of steam engines of the same power, and although the maximum pressures on them

may be as high in the one case as in the other, the mean pressure in the internal-combustion engine will be much less, as there is only one working stroke in four.

If we assume that the satisfactory working of a bearing depends upon the product of pressure and velocity not exceeding a safe limit, it will follow that the lower the pressure the higher the speed at which the bearing can be run, and this rule will be found to be generally observed when we compare steam and internal-combustion engines used for such work on shore, where the design has become standardized to the requirements. For example, a certain 10-horsepower (nominal) steam engine runs at 135 revolutions per minute, while a 20-brake-horsepower oil engine, which is capable of doing the same work, runs at 245 revolutions.

Taking the three factors, lighter accessories, lighter parts, and higher possible speeds of revolution, into consideration, it would appear that the figure 1/3.2, which was arrived at from considerations of maximum and mean pressures alone, can be considerably increased. How much it may be increased is more or less a matter of conjecture, but it will possibly be of the order of 1/1.5 to 1/2.5, so that for engines of the same linear dimensions the horsepower per ton will in the internal-combustion engine only be about one-half what it will be in a steam engine.

So far we have left out of account the question of weight of boilers and their accessories. These will have a counterpart in the gas producers for the gas engine, but have no equivalent in the oil engine. So far as the gas engine is concerned, the weight and space required for producers is largely dependent upon the type of fuel used and the cleaning arrangements necessary to deal with the gas, and this is too large a subject for the present paper. As the weight of boilers is usually about equal to the weight of engines, it follows from the above conclusion that the horsepower per ton will, for the complete installation, be about equal to that of an oil engine of the same linear dimensions as the steam engine. It is possible that there may be some saving, but it appears certain that if the internal-

combustion engine is to develop on lines parallel to that of the steam engine—that is, with few cylinders—there will be no very great saving, such as has sometimes been imagined by inference from the results obtained with small-sized units, where the speed of revolution is high.

The limit of speed of revolution will generally be determined by the inertia forces, and these have to be considered first as separate forces requiring adequate strength of the individual parts; and, secondly, in combination with a view of making their algebraic sum as nearly zero as possible. With a sufficient number of cylinders complete balance is comparatively easy to obtain, but the individual forces still remain and must be provided for. It is usual to assume that the inertia forces, expressed in pounds per square inch of piston area, should be less than the compression pressure, in order to avoid shock due to reversal of stress when combustion follows at the end of compression. It is argued that there is no reversal of stress on the idle strokes (suction and exhaust), but this is only true if we neglect the effect of friction of the piston and slackness in the bearings. It is well to remember, when considering the possibility of shock, that the effect of a load suddenly applied is double that of a load steadily applied, and further, that the calculation for inertia force can only be made on the assumption of uniform angular velocity of the crank, an assumption probably some distance from the truth in any reciprocating engine. The safe rule is to keep the calculated inertia force per square inch of piston as low as practicable; and if we assign a limit to this which, from present experience, the author would be inclined to put at 100 pounds, we obtain a formula of the following kind:

$$w \,.\, l \,.\, N^2 \text{ constant},$$

where w is the weight in pounds of the reciprocating parts per square inch of piston area, l is the stroke, and N the number of revolutions per minute This may also be written—

$$w \,.\, P \,.\, N = \text{constant},$$

where P is the piston speed; from which it follows that, at constant piston speed, the permissible speed of revolution varies inversely as the weight of reciprocating parts per square inch of piston. The permissible speed of revolution can be connected with the diameter of cylinder in the following manner: If we look at the table of weights given above we find that the reciprocating weights per square inch of piston increase with the diameter, and are approximately proportional thereto. This is reasonable; for, if we take one part—say the base of a piston, for example—its weight will vary as its area and thickness; and as for equal strength the thickness will vary as the diameter, the weight will vary as the cube of the diameter; hence, per square inch of piston, the weight will vary as the diameter. It will be found that similar rules hold good for other reciprocating parts. Hence, if as found above—

$$w \,.\, N = \text{constant, for constant piston speed,}$$

and if w varies as d, the diameter of cylinder, then—

$$d \,.\, N = \text{constant,}$$

that is, for a given piston speed the permissible speed of revolution varies inversely as the diameter. It follows, therefore, that a necessary condition of high speed of revolution and higher power per ton of weight is a small diameter of cylinder.

This reasoning is applicable to engines designed for a particular maximum pressure, but it can be extended to varying pressures by considering that the weights per square inch of piston will vary approximately as the pressures. If H be the maximum pressure, we may write—

$$d \,.\, N \,.\, H = \text{constant,}$$

that is, the higher the maximum pressure the lower the permissible speed of revolution with a given diameter of cylinder.

This relationship may be called a mechanical law of comparison, and may be applied approximately to determine the relative weights of engines of different dimensions. If we suppose, for example, a petrol engine of, say, 5-inch cylinder

diameter and 5-inch stroke, and at 1,200 revolutions to give 100 horsepower per ton, and if we take such an engine as a model for one ten times the linear dimensions, the diameter will be 40 inches, the stroke 50 inches, the allowable revolutions will be 120, and the horsepower per ton will be 10.

A modern marine steam engine having these dimensions for its high-pressure cylinder will give (engine weight only) about 20 horsepower per ton, or double the horsepower per ton of the internal-combustion engine of the same linear dimensions —a conclusion arrived at previously by a different method of reasoning.

The direction in which light weight is to be sought lies in keeping down the diameter of the cylinder, thus allowing a higher speed of revolution; but for large powers this involves a large number of cylinders.

That a larger number of cylinders than usual with steam practice is inevitable is apparent from considerations of uniformity of turning moment, but how far the number can be advisedly increased can only be determined by experience. Up to the present sixteen cylinders have been used on one shaft, and there seems no reason why this number should not be increased by successive steps of four or more cylinders at each step. The powers, however, which have been obtained per cylinder in engines whose design admits of application to propulsion are not large—probably not exceeding 100 horsepower; and until the unit is largely increased the very large powers required in many present-day vessels are out of reach of the internal-combustion engine, the immediate application of which would appear to be concerned mainly in the propulsion of boats and small vessels. In powers, say, of 500 horsepower on one shaft, it appears reasonable to expect 12 to 15 horsepower per ton of machinery weight; and though this is low compared with larger-powered engines for naval work, it is higher than the ordinary run of merchant-service practice. The figures given above refer to a complete installation of oil engines without auxiliaries; the weights will be greater with gas engines, for the reasons already given.

In connection with the use of a large number of cylinders some reference should be made to the proposals which have been made from time to time for transmitting power from engine to propeller electrically.

By this means the total power can be split up into a number of convenient units, and a much larger number of cylinders can be employed than is immediately practicable to place together on one or two shafts with direct coupling to the propellers. The weight of the engine per horsepower is thus kept low, but, on the other hand, the extra weight due to the addition of dynamos and motors to the installation will counteract the saving due to the large number of cylinders of small diameter. There are the further disadvantages of increased cost and higher fuel consumption per shaft horsepower. On the other hand, the system appears to be the only one at present practicable by which the internal-combustion engine can be applied to the propulsion of vessels requiring several thousand horsepower.

The author has made an approximate estimate of the weight of such an installation for 3,000 horsepower. The total power is supposed generated by ten equal-sized electric sets of 300 horsepower, or, say, 200 kilowatts each, running in parallel with one another, and the vessel propelled by three shafts, each fitted with a 1,000-horsepower variable-speed reversing motor. The weight of the generating sets would be approximately 240 tons, and the weight of the three motors, with bedplates, 43 tons. Switchboards, cables, shafting, and propellers would probably require not less than 17 tons, making the total weight about, but not less than, 300 tons; and of this weight, approximately one-third would be due to the electrical portion. This weight is exclusive of auxiliary machinery, the installation of which in a large vessel propelled by internal-combustion engines would give rise to a number of problems of some importance, particularly with regard to steering apparatus.

The total floor space required for the generating sets would be about 1,500 square feet, or 2 horsepower per square foot. This is probably somewhat better than the results obtained

with steam engines and cylindrical boilers as fitted in the mercantile marine; but it does not compare favorably with the space required for engines and water-tube boilers as used in naval work. The problem of electric propulsion is one of great interest, and its adoption in special circumstances may be justified; but it would seem that generally it does not offer sufficient commercial advantages for mercantile work, and, on the other hand, for naval work it cannot compete with existing steam machinery in the all-important considerations of weight and space. The question of the economical application of the internal-combustion engine to propulsion in large sizes, whether the economy is viewed from the standpoint of cost, weight or space, is largely bound up with the problem of making a reliable reversing engine. In small sizes, the familiar devices of reversing propeller or reversing clutch will, no doubt, do all that is required, but few, if any, marine engineers would care to contemplate their adoption in thousands or even several hundreds of horsepower.

An account of a successful experiment in a reversing engine of 80 brake horsepower of the Hornsby type has been given by the author in another place.* The method there described is applicable to larger engines of this type. With such a reversing engine there is a possibility of considerable saving in space over steam machinery, and as compared with mercantile practice some appreciable saving in weight without reference to saving in quantity of fuel carried.

The main claims, however, which the internal-combustion engine has for consideration are in the superior fuel economy and smaller number of men required for working, and these claims are so unquestionable that there appears no reasonable doubt that it must eventually, in a large number of cases, take the place of steam machinery at sea, as it is now doing on land.

The author has endeavored to place the question of weight and space required for internal-combustion engines in its true perspective, and although the saving, particularly in space, will in most cases be real, he would repeat that, on the whole, there

* Proceedings of the Institution of Civil Engineers, Vol. CLXVIII.

will be no great saving in weight such as has sometimes been imagined by inference from results obtained in small units.

The author is indebted to Mr. W. H. Scott, of Messrs. Laurence, Scott & Co., for the weight of the dynamos and motors embodied in the estimate of total weight of the arrangement of electric propelling apparatus described in the text.—"Engineering."

FORMULAE CONNECTING THE PRESSURE AND TEMPERATURE OF SATURATED STEAM.

By S. Godbeer.

An empirical formula giving the relation between the pressure and temperature of saturated steam, to be satsfactory, must (*a*) cover the whole range, (*b*) give a fair representation of those experimental results which probably approximate most closely to the true relation, and (*c*) be easy of calculation. None of the formulæ in common use satisfy these conditions, and with a view of supplying the deficiency a new formula is here given.

In the first place, however, some general remarks are necessary on the subject of the data available. Experiments by Thiesen and Scheel, Wüllner and Grotrian, Cailletet and Colardeau, Ramsay and Young, Knoblauch, Linde and Klebe, and Holborn and Henning will be referred to subsequently, and for the sake of brevity only the first name in each case will be quoted.

In a paper by Henning, published in the "Annalen der Physik," dated April 4, 1907, all the best experimental results up to that date are tabulated and discussed, and all necessary references given to the sources from which the observations have been collected. A study of the original papers of the various observers sufficiently indicates that the experimental difficulties are very great, and, therefore, the constants adopted in any empirical formula must be regarded as liable to revision in the light of future observations.

The latest experiments are those of Holborn, ranging from

50 degrees C. to 205 degrees C., given in the "Annalen der Physik" of August 25, 1908; and with this paper is a table showing the pressure for every degree from 0 degrees C. to 205 degrees C., based on the author's observations and on those of Thiesen from 0 degrees C. to 25 degrees C. This table has been selected as giving the most reliable values, so far as it extends, for the reasons given below.

As is well known, the subject under consideration was in such confusion in the early part of last century that, in 1846, Regnault undertook his famous experiments with a view of establishing the relation between pressure and temperature with such accuracy as to leave no need for subsequent correction, and from 50 degrees C. to 230 degrees C., over which range the same apparatus was used, his results appear to be such as he aimed at, being in close agreement with those of Holborn referred to above, and those of Knoblauch above atmospheric pressure, and those of Wüllner and Wiebe below 100 degrees C.

The important experiments of Ramsay must also be referred to. The containing vessel used was a glass tube of small dimensions, and the pressure observed at any given temperature varied with the amount of water in the tube, due apparently to the vapor being attracted by the glass surface it was in contact with, which varied according to the amount of liquid experimented with. This explains why the observed pressures are lower than Regnault's, and also why they are not suitable for the purpose now under consideration. Ramsay's paper, however, deserves careful study, especially for the remarks on various formulæ, and the statement contained therein that the vapor pressure of ice is not continuous with that of water should be borne in mind.

Thiesen made careful experiments from 0 degrees C. to 25 degrees C. with results which are well supported by those of Marvin, and, as embodied in Holborn's table referred to above, these have been adopted as the most reliable at present available. It may be remarked, however, that any curve through Thiesen's results which would accurately joint up to a curve through Holborn's results above 50 degrees C. would probably

require a complicated formula for its expression, and further experiments in these lower ranges of temperature would be welcome. For the present, however, if greater accuracy below 50 degrees C. is required than that given by the general formula, the following may be used as fairly representing Thiesen's records, pressures being in millimeters of mercury and temperatures on the Centigrade scale:

$$\log (p + 0.12) = 8.52772 - \frac{1{,}992.21}{t + 253.6},$$

from which the following table has been calculated:

t	p Thiesen.	p Formula.
0	4.5790	4.5790
14.5679	12.4385	12.4333
15.0593	12.8285	12.8323
16.3603	13.9189	13.9431
19.8402	17.3622	17.3381
19.8438	17.3411	17.3420
24.9749	23.6818	23.6642
25,4748	24.3308	24.3763
50	—	92.30

This formula for pressures in pounds per square inch and temperatures on the Fahrenheit scale becomes

$$\log (p + 0.00232) = 6.8142237 - \frac{3{,}585.978}{t + 424.48}.$$

From 230 degrees C. up to the critical temperature there are three sets of observations—viz: those of Knipp, Battelli and Cailletet, differing widely each from the others, so that it becomes necessary to select one as a basis for the general formula. Now all the formulæ which best represent Regnault's results, such as his own, give pressures at the highest temperatures much lower than those in either of the three sets above mentioned, which seems to be a reason for choosing the lowest set unless other considerations outweigh this. Knipp's pressures

are the highest of all, and on referring to his paper describing his apparatus and enumerating the difficulties he encountered with leaks, calibrations for temperatures, doubt as to the purity of the water used, etc., its seems that this set of observations should be discarded.

Battelli's method was to trace the isothermals and note the pressures at which they became horizontal straight lines. This does not seem likely to give such accurate results as the statical method adopted by Cailletet. The diagrams of the isothermals with the observation points spotted on show the latter to be sufficiently far off the curves to leave doubt as to the value of the deduced saturation pressures. Moreover, the pressures he finds for various temperatures will not lie at all fairly on any curve through Regnault's records that would seem to be admissible, and these pressures at the higher temperatures are, on the whole, above those of Cailletet.

For these reasons Battelli's observations are also rejected.

There only remains the experiments of Cailletet. A perusal of the paper in which they are given shows that these were conducted with great care and skill, and that there is no reason to doubt the substantial accuracy of the results obtained, which have therefore been used in the construction of the general formula, and here it may be observed that there are some irregularities in Cailletet's pressures at points of maximum and minimum deviation from the formula which it is desirable to examine.

For example, at 245 degrees C. the observed pressure is 35.5 atmospheres. Using the formula

$$\log p = 0.007738\, t - 0.34118,$$

the following results are obtained:

t.	230	235	240	245	250	255	260
p calculated...	27.451	30.01	32.81	35.86	39.20	42.86	46.849
p experiment.	27.5	30.0	32.8	...	39.2	42.9	46.8

So that to be in regular order with the adjacent observations the pressure at 245 degrees C. should be 35.9 atmospheres. Again, using the formula

$$\log\ (p+26) = 0.00466\, t + 0.65055,$$

the following results are obtained:

t.	260	265	270	275	280	285	290	295
p calculated...	46.80	50.82	55.049	59.52	64.23	69.20	74.45002	79.99
p experiment.	46.8	50.8	55.0	59.4	64.3	69.2	74.5	80.0

which indicates that the pressures recorded at 275 degrees and at 280 degrees require to be modified to 59.5 and 64.2 atmospheres to give a regular series.

Finally, from the formula

$$\log\ (p+10) = 0.005534\, t + 0.32179$$

the following results are obtained:

t.	285	290	295	300	305	310	315
p calculated...	69.246	74.46	80.02	85.94	92.249	98.98	106.14
p experiment.	69.2	74.5	80.0	86.2	92.2	99.0	106.1

showing that, to be in regular order, the pressure at 300 degrees C. is 85.9 atmospheres.

This last correction has a special importance, inasmuch as Cailletet apparently uses the value 86.2 to determine the constants in formulæ due to Bertrand and Clausius, which he employs to confirm his experimental results.

If pressure and temperature curves for these three sets of records above 230 degrees C. be drawn, it becomes apparent that Cailletet, Battelli and, much less obviously, Knipp, all find

a sudden disturbance in the general trend of the curve between 240 degrees and 270 degrees C. Further experiments in this region of temperatures would be interesting.

With these preliminary remarks the general formula mentioned previously may be stated as follows. For pressures in millimeters of mercury, and temperatures on the Centrigrade scale:

$$\log p = \frac{1319\ (t + 226)\ (t + 2299)^2}{192028\ (t + 808)\ (t + 329)} - 30.203 \qquad \text{(i.)}$$

For pressures in pounds per square inch and temperatures on the Fahrenheit scale

$$\log p = \frac{887\ (t + 374)\ (t + 4107)^2}{233893\ (t + 1422)\ (t + 560)} - 31.695365 \qquad \text{(ii.)}$$

Next putting

$$\frac{100\ (\text{observed pressure} - \text{calculated pressure})}{\text{calculated pressure}} = q,$$

it will be found that the maximum numerical value of q is as given in Table I.

TABLE I.

Temperature, ° C.	Temperature, ° F.	q Formula (i).	q Formula (ii).
0	32	−0.170	−0.179
15	59	+0.172	+0.178
240	464	−0.837	−0.791
265	509	+0.718	+0.780
305	581	−0.053	+0.053
330	626	+0.121	+0.247
365	689	−0.898	−0.729

The pressures for various temperatures, as calculated from formula (i.) and Regnault's formula K, are given in Table II with the selected observations.

TABLE II.

Temp., ° C.	p Formula (i).	p Observed.	p Regnault K.	q	Temp., ° C.	p Formula (i).	p Observed.	p Regnault K.	q
0	4.587	4.579	...	−.170	190	9,404	9,405	9,407	−.004
5	6.538	6.541	...	+.043	200	11,652	11,647	11,656	−.043
10	9.192	9.205	...	+.137	210	14,299	...	14,302	...
15	12.757	12.779	...	+.172	220	17,395	...	17,390	...
20	17.49	17.51	...	+.137	225	19,126	19,076	19,114	−.261
25	23.69	23.69	...	...	230	20,989	20,900	20,965	−.424
30	31.73	31.71	...	−.070	235	22,991	22,800	22,950	−.831
40	55.18	55.13	...	−.092	240	25,138	24,928	25,074	−.837
50	92.33	92.30	...	−.036	250	29,901	29,792	29,765	−.364
60	149.18	149.19	...	+.007	260	35,340	35,568	35,086	+.645
70	233.49	233.53	...	+.017	265	38,333	38,608	37,999	+.718
80	355.0	355.10	...	+.028	270	41,521	41,800	41,088	+.672
90	525.71	525.80	...	+.017	280	48,517	48,792	...	+.567
100	760.0	760.0	760	+.007	290	56,410	56,620	...	+.372
110	1,074.6	1,074.5	1,074	−.009	300	65,276	65,284	...	+.012
120	1,489.3	1,488.9	1,487	−.027	310	75,210	75,240	...	+.040
130	2,026.0	2,025.6	2,023	−.020	320	86,309	86,412	...	+.119
140	2,710.0	2,709.5	2,706	−.018	330	98,681	98,800	...	+.121
150	3,568.7	3,568.7	3,564	...	340	112,441	112,252	...	−.168
160	4,633.0	4,633.0	4,629	+.006	350	127,716	127,300	...	−.326
170	5,936.0	5,937.0	5,933	+.020	360	144,636	143,564	...	−.741
180	7,513.0	7,514.0	7,513	+.011	365	153,761	152,380	141,241	−.898

As an example, the value of p when $t = 100$ is determined below from formula (i). In (A) the steps are fully written out, while in (B) only the essential portions are given.

(A)

$\text{Log}\,(100 + 226) = 2.5132176$
$\text{Log}\,(100 + 2299) = 3.3800302$
$\text{Log}\,(100 + 2299) = 3.3800302$

(N. B.—The first two figures of the difference are 1.4 below about 27 degrees C. and 1.5 above.)

$\text{Sum} = 9.2732780$

$\text{Difference} = 1.5196152 = \log 33.083785$

-30.203

$\text{Sum} = 7.7536628 \qquad 2.880785 = \log p$

$\text{Log}\,\frac{192028}{1319} = 2.1631197 \qquad \therefore p = 760 \text{ when } t = 100$

$\text{Log}\,(100 + 808) = 2.9580838$
$\text{Log}\,(100 + 329) = 2.6324573$

(B)

132176		
800302		
800302		
732780		
1.5196152	3.083785	$t = 100$
536628	0.203	$p = 760$
631197	2.880785	
580858		
324573		

This demonstrates the ease of application of the formula.—"Engineering."

LECTURES BY DR. ERNST J. BERG, DELIVERED TO EMPLOYEES OF THE COMMONWEALTH EDISON COMPANY, FEBRUARY NINETEENTH, 1910.

I.—*Notes on the Magnitude of the Losses Incidental to the Transformation of the Energy in Coal to Electrical Energy.*

The heat balance will be illustrated for a specific case, which will be that of a steam turbine station of 1,000 kw. capacity.

It will be assumed that the combined efficiency of the turbine and the generator is 65 per cent. and that the turbine is supplied with steam of 200 pounds absolute pressure at 150 degrees F. superheat, and that the condenser pressure is 96.6 per cent., corresponding to a 28.4-inch vacuum with a barometer pressure of 30 inches, or .75 pound absolute.

The available energy in foot-pounds in each pound of superheated steam when expanded adiabatically from a given pressure, and superheat to the pressure of the vacuum, is the difference between the total heat input and that which is left as liquid heat and latent heat in the mixture at the lower pressure.

Thus

$$E = 778\,[H_1 + C_p t_1 - (q_2 + x_2 r_2)], \quad . \quad . \quad . \quad (1)$$

where

H_1 = total heat of sat. steam at initial pressure p_1.
C_p = spec. heat of superheated steam.
t_1 = superheat in degrees F. at pressure p_1.
q_2 = heat of liquid at lower pressure p_2.
x_2 = quality of the steam at the pressure p_2.
r_2 = latent heat at the pressure p_2.

The steam tables enable us to insert all values in the above equation except the value of x_2. This quality is obtained from the fact that the entropy is constant before and after the expansion; that is, the mathematical expression entropy, which is the ratio of the heat received to the absolute temperature of reception is constant. The entropy of superheated steam is

$$= C_p \log_e \frac{T_1 - t_1}{T_1} + \frac{r_1}{T} + o_1. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The entropy of moist steam is

$$= \frac{x_2 r_2}{T_2} + o_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

thus

$$x_2 = \frac{T_2}{r_2}\left[C_p \log_e \frac{T_1 + t_1}{T_1} + \frac{r_1}{T_1} + o_1 - o_2\right]. \quad . \quad . \quad . \quad (4)$$

In the particular instance referred to above we have—

p_1	= 200 pounds absolute.	
p_2	= .75 pound absolute.	
H_1	= 1,193.3 (from steam tables).	
C_p assumed to be	= .5	
T_2	= 92.1 deg. F.	= 552.1 degrees abs.
T_1	= 381.6 deg. F.	= 841.6 degrees abs.
t_1	= 150 deg. F.	Superheat at p_1.
r_1	= 843.4	Latent heat at p_1
r_2	= 1,049.8	Latent heat at p_2.
o_1	= .5429	Entropy of the liquid.
o_2	= .1152	Entropy of the liquid.

thus

$$x_2 = \frac{552.1}{1{,}049.8}\left[.5\left(2.3\log_{10}\frac{841.6+150}{841}\right)+\frac{843.4}{841.6}+.5429 - .1152\right] = .8. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Substituting this value in the energy equation above, we get—

$$E = 778\,[1{,}198.3+.5\times 150-(60.2+.8\times 1{,}049.8)] = 290{,}000 \text{ foot-pounds.} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Since one kw. hour is equivalent to $\left(\frac{1{,}000}{746}\times 33{,}000\times 60\right)$ = 2,654,000 foot-pounds, the amount of steam consumed per kw. hour under these conditions will be $\frac{2{,}654{,}000}{290{,}000} = 9.15$ pounds, which is the theoretical water rate; but since the efficiency of the turbine is only 65 per cent., the real water rate will be—

$$\frac{9.15}{.65} = 14.1 \text{ pounds per kw. hour,} \quad . \quad . \quad . \quad (8)$$

Therefore the total flow for 1,000 kw. hour is 14,100 pounds per hour. The total heat in each pound of superheated steam, assuming $C_p = .5$, is $1{,}198.3 + .5 \times 150 = 1{,}273.3$ B.T.U. (9) Thus the total heat of 14,100 pounds is $14{,}100 \times 1{,}273.3 =$ 17,960,000 B.T.U. Since one kw. hour corresponds to 3,410 B.T.U. the energy converted to electricity. Therefore the energy not converted is 14,550,000, or 81 per cent. . (10)

Some of these losses are caused by the generator windage, copper and iron losses, and some by the bearings. These losses, which have nothing directly to do with the steam, amount to about 6 per cent. of the electrical output, or

$.06 \times 3{,}410{,}000 = 204{,}000$ B.T.U. . . . . (11)

The rest of the losses are found in the energy of the condensed steam, in the cooling water, in the friction loss in the turbine, in the heat convection and radiation from the turbine case, the condenser, piping, etc.

These losses thus amount to 14,550,000 — 204,600 = 14,345,400 B.T.U. . . . . (12)

The temperature of the condensed steam at 28.4 inches vacuum or .75 pound absolute pressure is as seen from steam tables, 92.5 degrees F. It is evident from our previous calculations that the exhaust is not superheated; it contains in reality about 8.6 per cent. moisture, the actual moisture being less than that corresponding to adiabatic expansion on account of the re-evaporation of a part of the moisture by the heat caused by the rotation loss of the turbine wheels.

Since the combined efficiency of the turbo unit, including the generator, is assumed at 65 per cent., and in all probability the generator has an efficiency of about 96 per cent., it is evident that the turbine efficiency is 67.8 per cent. The loss is largely all converted into heat; thus, while the available energy is 290,000 foot-pounds per pound of steam, 32.2 per cent. are partially wasted as heat, that is, 322 × 290,000 = 93,500 foot-pounds, or 120 B.T.U. Considering now, for simplicity's sake, that all work was done in one single pressure stage or turbine, then the pressure in the stage would be .75 pound. The latent heat at that pressure is 1,050 B.T.U.; thus, 120 B.T.U. evaporate 11.4 per cent. of the moisture, and thus we obtain the value of 20 per cent. — 11.4 per cent., or 8.6 per cent. as given above.

In returning the condensed steam to the boiler its temperature will drop slightly, due to conduction and radiation. This drop may be 7.5 degrees F. The returned liquid heat of the water corresponds therefore to a temperature of 85 degrees F. and is 53.06 B.T.U. Thus, the total heat returned in the feed water is 53.06 × 14,100 = 750,000 B.T.U. . (14)
The heat lost in feed water is obviously

$$7.5 \times 14{,}100 = 105{,}800 \text{ B.T.U.} \quad . \quad . \quad . \quad (15)$$

Since the condensed steam at 92.5 degrees F. contains as liquid heat 60.7 × 14,100 = 855,800 B.T.U., it is evident that the cooling water contains 14,345,400 —855,800 = 13,489,600 B.T.U.

As the steam goes from the boiler to the turbine there is a loss of heat which will be found as a drop in superheat. This might be 50 degrees F., so that the steam as it leaves the boiler has a superheat of 200 degrees F. This loss in heat then represents

$$50 \times .5 \times 14{,}100 = 352{,}500 \text{ B.T.U.} \quad . \quad . \quad (17)$$

The boiler must therefore supply

$$17{,}960{,}000 + 352{,}500 = 18{,}312{,}500 \text{ B.T.U.} \quad . \quad (18)$$

Through the return of the feed water, however, we get 750,000 B.T.U. Thus the necessary coal corresponds to 17,562,500 B.T.U.

At a boiler efficiency of 80 per cent. this means that

$$\frac{17{,}562{,}500}{.80} = 21{,}953{,}000 \text{ B.T.U.} \quad . \quad . \quad . \quad (20)$$

of coal must be supplied corresponding to an amount of 1,690 pounds per hour, or $\frac{21{,}953{,}000}{1{,}000 \times 13{,}000} = 1.69$ pounds of coal per kw. hour with coal containing 13,000 B.T.U. per pound. (21)

Heat balance.	B.T.U.	Per cent.
Boiler-plant loss, ...	4,390,500	20.0
High-pressure steam-pipe loss, . .	352,500	1.6
Rotation losses, gen., ...	204,600	.9
Electrical output, gen., ...	3,410,000	15.6
Cooling-water loss, ...	13,489,600	61.4
Heat loss in feed water, ...	105,800	.5
Total, ...	21,953,000	100.0

II.—*Notes on the Economy of Steam Heating by Independent Low-pressure Steam Boiler or from the Exhaust of the Steam Turbine.*

To get some practical basis for comparison it will be assumed that 1,000 kw. electric power is wanted and that this power can be obtained from the turbine even if it is operating

THEORETICAL WATER RATE FOR VARIOUS PRESSURES, VACUUM AND SUPERHEAT.

Superheat, degrees F.		200°	150°	100°	50°	0°
Pounds absolute pressure.	In vacuum.	Theoretical water rate.				
215	29	8.45	8.7	8.95	7.2	9.45
215	28	9.0	9.25	9.5	9.75	10.0
215	27	9.5	9.75	10.05	10.3	10.55
215	26	9.9	10.2	10.50	10.8	11.11
215	20	11.65	12.07	12.50	12.87	13.25
215	10	13.65	14.15	14.65	15.15	15 65
215	0	15.25	15.8	16.4	16.9	17.4
215	2-lb. bk. pres.	15.05	16.07	17.1	17.6	18.2
200	29	8.55	8.8	9.05	9.32	9.6
200	28	9.1	9.4	9.7	9.95	10.2
200	27	9.6	9.9	10.2	10.45	10.75
200	26	10.0	10.32	10.65	10.97	11.3
200	20	11.75	12.4	12.8	13.2	13.6
200	10	14.0	14.5	15.05	15.57	16.0
200	0	15.6	16 2	16.9	17.4	17.9
200	2-lb. bk. pres.	16.3	16.9	17.5	18.0	18.7
175	29	8.75	9.02	9.3	9.55	9.8
175	28	9.35	9.62	9.9	10.18	10.45
175	27	9.85	10.15	10.45	10.75	11 05
175	26	10.3	10.65	11.0	11.3	11.6
175	20	12.4	12.82	13.25	13.65	14.05
175	10	14.65	15.77	15.70	16.25	16 8
175	0	16.4	17.35	18.3	18.6	18.9
175	2-lb. bk. pres.	17.1	17.75	18.4	19.1	12,8
165	29	8.85	9.12	9.4	9.63	9.86
165	28	9.45	9.72	10.0	10.3	10.6
165	27	10.0	10.32	10.65	11.0	11.3
165	26	10.5	10.85	11.2	11.55	11.9
165	20	12.6	13.05	13.5	13.9	14.3
165	10	14.9	13.05	16.1	16.57	17.05
165	0	16.85	17.52	18.2	18.75	19.3
165	2-lb. bk. pres.	17.6	17.3	19.0	...	20.3
140	29	9.12	9 37	9.65	9.9	10.2
140	28	9.75	10.05	10.35	10.65	10.95
140	27	10.3	10.62	10.95	11.28	11.62
140	26	10 85	11.20	11.55	11.95	12.3
140	20	13 1	13.6	14.1	14.6	15.1
140	10	15 8	16.4	17.0	17.6	18.2
140	0	17.9	18.6	19.3	20.0	20.65
140	2-lb. bk. pres.	18 6	18 95	20.3	21.0	21.8
125	29	9.35	...	9.88	...	10.4
125	28	10 03	...	10.6	...	11.2
125	27	10.70	...	11.3	11.6	11.9
125	26	11.23	...	11 9	...	12.55
125	20	13 65	...	14.6	...	15.50
125	10	16 50	...	17.7	...	18.98
125	0	18 90	...	20.4	...	21.80
125	2-lb. bk. pres.	19.85	...	21.6	...	23.1
100	29	9.77	...	10.25	...	10.89
100	28	10.50	...	11.16	...	11.72
100	27	11.2	...	11.9	...	12.50
100	26	11.83	...	12.65	...	13.23
100	20	14.70	...	15.80	...	16.65
100	10	18.0	...	19.45	...	20.7
100	0	20.9	...	22.28	...	24.25
100	2-lb. bk. pres.	22.15	...	24.25	...	25.82
Atmosphere (14.7 lbs.)	29	15.4	...	16.65	...	17.7
Atmosphere (14.7 lbs.)	28	17.65	...	19.1	...	20.4
Atmosphere (14.7 lbs.)	27	20.0	...	21.8	...	23.3
Atmosphere (14.7 lbs.)	26	22.40	...	24.5	...	25.4
10-inch vacuum (10 lbs.)	29	17.25	...	18.8	...	20.2
	28	20.20	...	22.1	...	24.2
	27	22.50	...	25.8	...	28.6
	26	26.95	...	30.0	...	33.0

non-condensing with a back pressure of 10 pounds, that is, at an absolute pressure of 24.7 pounds. It will be assumed, as in the first case, that the initial pressure is 200 pounds absolute, that the superheat at the turbine is 150 degrees F. and that the combined efficiency of the turbine and generator is 64 per cent. (This efficiency is assumed lower than the first case, since the turbine losses due to rotation are greater). Referring to equations (1) and (4) we determine the following values:

$$
\begin{aligned}
p_1 &= 200. \\
p_2 &= 24.7. \\
H_1 &= 1{,}198.3. \\
C_p &= .5. \\
T_2 &= 239.5 + 460 = 699.5. \\
T_1 &= 381.6 + 460 = 841.6. \\
t_1 &= 150. \\
r_1 &= 843.4. \\
r_2 &= 946.5. \\
o_1 &= .5429. \\
o_2 &= .353.
\end{aligned}
$$

Substituting these values in (4) and (1) we get $x_2 = .94$, and $E = 137{,}600$ foot-pounds per pound of steam.

Thus the theoretical water rate is $\frac{2{,}654{,}000}{137{,}600} = 19.3$, and the actual water rate is $\frac{19.3}{.64} = 30.2$ pounds per hour.

With 1,000 kw. output the flow is thus 30,200 pounds per hour. The total heat input is $1{,}273.3 \times 30{,}200 = 38{,}454{,}000$ B.T.U. The electrical output is 3,410,000 B.T.U. Thus, the energy not converted to electrical energy is 35,044,000, or 91.2 per cent.

In carrying the investigation farther we will assume that the same percentage loss by various inefficiencies not connected with the steam part is 6 per cent. = 204,600 B.T.U. The remaining losses are thus 34,839,400 B.T.U. These are available for steam heating and heating of the feed water.

It may be assumed that the condensed steam returns from the radiators at a temperature of 85 degrees F. and that this water is fed to the boilers. The liquid heat at this temperature is 53.06; thus the heat returned is $53.06 \times 30{,}200 = 1{,}602{,}400$, and the heat used for heating is

$$34{,}839{,}400 - 1{,}602{,}400 = 33{,}237{,}000 \text{ B.T.U.}$$

To complete the heat balance we may assume, as before, a drop in temperature of the superheated steam of 50 degrees F., corresponding to 755,000 B.T.U. The boiler must thus produce 38,454,000 + 755,000 — the returned heat, or 37,606,600 B.T.U. With the boiler efficiency of 80 per cent. this corresponds to 47,008,200 B.T.U. from the coal, or with coal at 13,000 B.T.U. per pound, 3,616 pounds of coal per hour for 1,000 kw. electrical output and 33,237,000 B.T.U steam heating.

Heat balance.	B.T.U.	Per cent.
Loss in boiler,	9,401,600	20.0
Loss in steam pipes,	755,000	1.6
Loss in radiation, etc.,	204,600	.4
Electric energy,	3,410,000	7.2
Heating purposes,	33,237,000	70.8
Total,	47,008,200	100.0

If instead of using exhaust steam the turbine was running condensing, and a special low-pressure boiler of 24.7 pounds absolute pressure supplied the steam required for heating, we get the following heat balance, assuming, as in the previous case, that 33,237,000 B.T.U. are needed for steam heating, and that 1,000 kw. electric power is obtained from a turbine operating with an initial pressure of 200 pounds absolute, 28.4 inches vacuum and 150 degrees F. superheat. It will also be assumed that the condensed steam from the heating system is returned to the low-pressure boiler at 85 degrees F. We have, then, B.T.U. required for heating = 33,237,000, and B.T.U. required from coal at 80 per cent. efficiency of

boiler = 41,546,200, or with coal of 13,000 B.T.U. per pound we require 3,196 pounds per hour for heating purposes.

Heat balance for low-pressure boiler.	B.T.U.	Per cent.
Heating, ...	33,237,000	80
Loss in boiler, ...	8,309,200	20
Total, ...	41,546,200	100

The combined coal for one hour for heating by independent boiler and for furnishing 1,000 kw. from condensing turbine is thus 3,196 + 1,690 = 4,886 pounds per hour.

In comparing the two systems we have thus, for steam heating from turbine exhaust and for 1,000 kw. electrical energy delivered, **3,616 pounds** of coal per hour, and for steam heating from independent boiler and 1,000 kw. supplied by separate condensing turbine **4,886 pounds** of coal per hour. In other words, there is a saving by the first method of 1,270 pounds, or 26 per cent. Incidentally, it is necessary to notice that if all the coal required for steam heating in the second case is charged up for heating in the first case, we find that the coal required for electric power is only 3,616 — 3,196 = 420 pounds, or .42 pound per kw. hour, or 24.8 per cent. of the coal required in the generation of electric power, even under the most favorable conditions of superheat and vacuum.

The calculations emphasize the desirability of doing steam heating from the exhaust steam, and show numerically the magnitude of the gain.

III.—*The Effect of Superheat, Vacuum, Initial Pressure and Feedwater Temperature on the Water Rate and Coal Consumption of Turbines.*

By referring to the equations given in last lecture, or the tabulation of theoretical water rates distributed at the time, the theoretical water rate can be determined for any values of initial pressure, superheat and vacuum.

So, for instance, with 200 pounds absolute initial pressure,

dry saturated steam and 28 inches vacuum, the theoretical water rate is 10.2 pounds per kw. hour. At 29 inches vacuum it is 9.6 pounds per kw. hour, and at 27 inches vacuum it is 10.75 pounds; thus, using 28 inches vacuum as a basis for comparison, there is a decrease in water rate of 5.9 per cent. when increasing the vacuum 1 inch, and there is an increase in water rate of 5.4 per cent. when lowering the vacuum by 1 inch.

Thus, on an average, we conclude that around 28 inches vacuum each inch of vacuum changes the water rate about 5.7 per cent.

With steam at 200 degrees superheat, the theoretical water rate at 28 inches vacuum is 9.1; at 29 inches vacuum it is 8.55, and at 27 inches vacuum it is 9.6; thus the average change in water rate per inch of vacuum is also 5.7 per cent.

Thus in general we can state that, disregarding the condition of the steam—at about 200 pounds absolute initial pressure and about 28 inches vacuum—there is a change of water rate per inch of vacuum of about 5.7 per cent. With lower initial pressure, as, for instance, 140 pounds absolute, the average change of water rate is 6.4 per cent. per inch of vacuum, thus considerably greater than at the higher initial pressure, as could be expected.

The effect of superheat on the water rate can also be determined from the equation of available energy or by referring to the same table.

So, for instance, at 200 pounds absolute pressure and 28 inches vacuum the theoretical water rate is 10.2 pounds with saturated steam. With steam at 200 degrees F. superheat it is 9.1 pounds; thus the decrease in water rate is 10.8 per cent., or each per cent. decrease in water rate requires 18.5 degrees superheat. This value remains practically the same, whether the vacuum is 27 inches or 29 inches. It is also practically the same for all commercial initial pressures.

It is an interesting fact, however, that while the gain in water rate by vacuum is practically that demanded by theory, the gain by superheat is almost 50 per cent. greater. The

fact that the gain by vacuum is as much as theory demands is interesting, since it indicates that the operation of the lower stages must be very efficient with high available energies and corresponding high steam speeds, since the lower or, indeed, the last stage is practically the only one affected by the change of vacuum, and may with high vacuum convert far more energy than each of the upper stages.

The large gain by superheat is no doubt due to the fact that the rotation loss is less than with initially-saturated steam. It has been found that this loss is a function of the quality of the steam. A mixture of steam with a considerable amount of water acts to some extent as a water brake.

It must, of course, not be forgotten that high vacuum is only possible with relatively low temperature of the cooling water, and that the amount of cooling water is necessarily considerably greater than with low vacuum. The condenser itself is also considerably larger than for lower vacuum. However, the saving by reduced boiler installation more than compensates for the costs of the larger circulating pumps and condenser, etc.

The great saving by the use of high vacuum is, however, in the coal consumption. The per cent. saving in coal corresponds practically to the gain by increased vacuum in the water rate. So, for instance, assuming the same efficiency of the turbine when operating at 27 inches and 28.5 inches vacuum the gain in water rate at average commercial initial pressure is 9 per cent., and 9 per cent. in water means substantially a saving of 9 per cent. in coal.

The gain in water rate by superheat only slightly affects the coal consumption. As an instance, will be determined the relative coal consumption for a turbine operated at 200 pounds absolute initial pressure with dry saturated steam and 28 inches vacuum, and another with same pressures, but 100 degrees superheat.

The total heat per pound of saturated steam at 200 pounds' pressure is 1,198.3, the heat corresponding to 100 degrees

superheat (assuming C_p as .5) is 50; thus the increase in heat with superheat is

$$\frac{50}{1{,}198.3} = .0417, \text{ or } 4.17 \text{ per cent.}$$

The available energy in the superheated steam between 200 pounds and 1 pound is 275,200 foot-pounds. In saturated steam it is 261,000 foot-pounds; thus, the increase in available energy is 5.37 per cent.; thus, the gain in coal is 5.37 per cent. — 4.17 per cent. = 1.2 per cent. As stated above, the gain in water rate is considerably greater than corresponds to the gain in available energy and is about 8 per cent.; thus, in practice the gain in coal by 100 degrees superheat is 8 — 4.17, or 3.83 per cent. In this calculation it has been assumed that the flue gases leave at the same temperature and that the thermal efficiency of the boiler in one case and boiler with superheater in the other is the same. In all probability this is not quite the case; nevertheless, there is a decided gain.

The initial pressure has some importance in connection with economy, but the gain by increasing the initial pressure is very small compared with that derived by a few inches of vacuum.

Referring again to the tabulation of the theoretical water rates:

At 200 pounds' pressure, saturated steam and 28 inches vacuum, the theoretical water rate is 10.2 pounds per kw. hour. At 175 pounds absolute and same vacuum it is 10.45 pounds; thus, the gain in water rate by raising the initial pressure 25 pounds is .25 pound, or 2.5 per cent. for 25 pounds, or 1 per cent. for each 10 pounds.

Effect of Feed-water Temperature on Steam Economy.—It is generally recognized that it is well to supply steam boilers with water of as high temperature as possible. There are many good reasons for this practice outside of the gain in economy, but it is well worth doing on that score alone.

To illustrate this a few typical cases will be considered.

1. The case when there is no feed water heating, which

might be the case in a plant using electrically-driven auxiliaries only.

2. The exhaust from the auxiliaries is used to heat the feed water.

3. The feed water will be heated partly by live steam and partly by the exhaust from the auxiliaries.

4. The feed water is heated partly by the exhaust and partly from a turbine which is "bled."

1. No Feed-water Heating.—We will assume a plant of 540 kw. capacity of main turbines and that the auxiliaries require 40 kw. Thus, the useful output is 500 kw.

It shall be assumed that the turbine is supplied with dry saturated steam at 165 pounds absolute pressure, that the vacuum is 28 inches and the water rate 19 pounds per kw. hour.

The flow per hour is then $19 \times 540 = 10{,}250$ pounds. At 28 inches vacuum the temperature of the condensed steam is 102 degrees F., at the boiler pressure it is 366 degrees F. Thus we need to supply $366 - 102 = 264$ B.T.U. as liquid heat per pound of water. The latent heat at 165 pounds' pressure is 855.6; thus, each pound of the flow requires $855.6 + 264 = 1{,}119.6$ B.T.U. and the total heat required is

$$1{,}119.6 \times 10{,}250 = 11{,}500{,}000 \text{ B.T.U.}$$

2. Engines or Turbines driving the Auxiliaries and their Exhaust (at Atmospheric Pressure) used to Heat the Feed Water.—As in the first case, it will be assumed that the useful output is 500 kw. at a water rate of 19 pounds. It will be of interest to determine the water rate of the small engines or turbines which at 40 kw. output would give sufficient exhaust steam at atmospheric pressure to raise the temperature of the feed water to 212 degrees F. Since the temperature of the condensed steam from the main turbine is 102 degrees F., we require $212 - 102 = 110$ B.T.U. per pound of steam; thus, since with 500 kw. output at a water rate of 19 pounds the flow is 9,500, the heat required is $9{,}500 \times 110 = 1{,}045{,}000$ B.T.U.

Assuming that the exhaust from the auxiliary engine contains 4 per cent. moisture, the heat input is the latent heat of 96 per cent. of the flow.

Let f be the flow from the auxiliary engine;

a the latent heat at atmospheric pressure, which is 965.8.

We have, then,

$$965.8 \times .96f = \text{heat input} = 1{,}045{,}000.$$

Thus the flow $f = 1{,}130$ pounds, which corresponds to a water rate of the auxiliaries of $1{,}130 \div 40 = 28.2$ pounds per kw. hour.

This water rate is far better than could be expected from small non-condensing turbines or engines; thus the heat in the exhaust is more than enough to bring the temperature of the feed water to 212 degrees F.

Some of the steam would have to exhaust into the atmosphere and would be lost.

If the water rate were 40 pounds per kw. hour, which is likely, then the amount of surplus steam would be $40\ (40 - 28.2) = 470$ pounds. This steam would have to be supplied from the source of water which might have a temperature of 60 degrees F., but before entering the boiler it could be heated to 212 degrees F.

Thus the total heat necessary to be supplied is:

For that part of the water which has to be drawn for the water supply,

Liquid heat $= 470\,(366 - 212) =$	72,400 B.T.U.
Latent heat $= 470 \times 855.6$	$= 403{,}000$
	475,400
For the rest of the flow $(9{,}500 + 1{,}130)$	$= 10{,}630$ lbs.,
Liquid heat $= 10{,}630\,(366 - 212)$	$= 1{,}638{,}000$
Latent heat $= 10{,}630 \times 855.6$	$= 9{,}110{,}000$
	10,748,000

Thus total heat $= 11{,}223{,}400$.

In this case, then, when a considerable portion of the exhaust steam is wasted, the gain is only slight, it being only about 2.4 per cent.

In large stations, where the amount of power taken by the auxiliaries is a much smaller percentage of the useful power, the gain is very considerable.

So, for instance, if the auxiliaries required only 5.6 per cent. of the useful power instead of 8 per cent., as was the case above, then no exhaust steam would go to waste and the total heat required would have been 10,748,000 B.T.U.

The saving in coal, then, would have been 6 per cent.

In very large stations the amount of exhaust steam is frequently insufficient to raise the temperature of the feed water to 212 degrees. It is then of interest to inquire whether any economy would result if live steam was delivered to the feed or if steam at substantially atmospheric pressure was drawn from the turbine.

As an illustration, it will be assumed that the auxiliaries require only 4 per cent. of the main flow, that the water rate of the main turbines is 15 pounds per kw. hour, and that the vacuum is 29 inches.

For the sake of simplicity we shall make the calculations for each 1,000 kw. of output of the main turbines. The temperature of the condensed steam is 80 degrees F., and the amount of condensed steam 15,000 pounds per hour.

The mixture of the condensed steam and the exhaust will take a certain temperature t, which is higher than 80 degrees and lower than 212 degrees.

The heat required is the liquid heat between temperatures t and 80 degrees F.; thus heat required is $15{,}000\,(t - 80) = 15{,}000\,t - 1{,}200{,}000$. The amount of the exhaust steam is $.04 \times 15{,}000 = 600$ pounds. Thus the heat given by the exhaust is:

Its liquid heat $= 600\,(212 - t) = 127{,}200 - 600\,t$, and its latent heat $= .96 \times 600 \times 965.8 = 557{,}000$.

Thus $15{,}000\,t - 1{,}200{,}000 = 127{,}200 - 600\,t + 557{,}000$; thus $t = 121$ degrees.

The heat required is:

Liquid heat for a flow of
$15{,}600 = 15{,}600\,(366 - 121) = 3{,}820{,}000$

Latent heat for a flow of
$15{,}600 = 15{,}600 \times 855.6 = 13{,}350{,}000$

$17{,}170{,}000$

3. Feed Water Heated Partly by Live Steam.—To raise the temperature of the feed water to 212 degrees we need, obviously, $15{,}600\,(212 - 121) = 1{,}420{,}000$ B.T.U.

With p pounds of live steam supply we get from the latent heat, ... $855.6\,p$ B.T.U.

From the liquid heat $p\,(366 - 212) = 154\,p$

Thus a total amount of heat of . . $1{,}009.6\,p$.

Thus $1{,}009.6\,p = 1{,}420{,}000$, or $p = 1{,}410$ pounds per hour. In this case, then, the total flow will be

$$15{,}000 + 600 + 1{,}410 = 17{,}010 \text{ pounds,}$$

and the heat required:

Liquid heat, $17{,}100\,(366 - 212) = 2{,}620{,}000$
Latent heat, $17{,}100 \times 855.6 = 14{,}550{,}000$

$17{,}170{,}000$

Thus there is no gain or loss except in so far as it is better to supply the boiler with the hotter water.

4. The Feed Water is Heated by the Exhaust of the Auxiliaries and also from the Turbine, Steam being drawn from the Turbine at Substantially Atmospheric Pressure.—As was shown above, to raise the feed-water temperature to 212 degrees we require 1,420,000 B.T.U.

The second stage of the turbines usually has about atmospheric pressure. Thus, since the latent heat at atmospheric pressure is 965.8 B.T.U., we require

$$\frac{1{,}420{,}000}{965.8} = 1{,}470 \text{ pounds of steam.}$$

At atmospheric pressure, or since the steam probably contains 4 per cent. moisture $\frac{1,470}{.96} = 1,530$ pounds per hour.

This steam has done some work in the main turbine, the amount of work being governed by the ratio of the available energy in the steam between the initial pressure and atmospheric pressure to the total available energy. As can readily be calculated from the equation for the available energy, and as can be seen from the tabulation of theoretical water rate, the available energy in the steam between the boiler pressure and the atmospheric pressure is approximately one-half of the total if it had expanded to 29 inches vacuum.

Thus the water rate for that part of the steam which passes through the upper part of the turbine only, and which is exhausted in the feed water is, say, $\frac{15}{.5} = 30$ pounds per kw. hour. The output of 1,530 pounds is, therefore,

$$\frac{1,530}{30} = 51 \text{ kw.}$$

The steam which goes through the entire turbine need, therefore, give only 949 kw.

Thus the total flow of steam is:

Main turbine, in its entirety, 949×15	=	14,240
Main turbine, non-condensing part,	=	1,530
Flow from auxiliaries,	=	600
		16,370 lbs.

Thus the heat required is:

Liquid heat, $16,370\,(366 - 212)$	=	2,500,000
Latent heat, $16,370 \times 855.6$	=	14,000,000
Total heat, . . .		16,500,000

The saving effected by bleeding the turbine is thus $17,170,000 - 16,500,000 = 670,000$ B.T.U., or 3.9 per cent.

LUBRICATION AND LUBRICANTS.

PROF. CHARLES F. MABERY, "Iron Age," Jan. 20, 1910.

Next to the conservation of the world's fuel supply there is probably no subject of greater importance in the manufacturing world than the control of waste power caused by imperfect lubrication and needless friction. Notwithstanding the increasing interest in more economical methods the immense losses from this source are scarcely appreciated. Writing on this subject, Archbutt states that 40 to 80 per cent. of the 10,000,000 H.P. used in Great Britain is spent in overcoming friction, and much of this is lost by imperfect or faulty lubrication. The conditions are probably worse in the United States on account of the cheapness of fuel, but it is safe to say that from 10 to 50 per cent. of the power employed is lost. Experts frequently find a loss of 50 per cent. or more in large factories spending thousands of dollars annually for lubrication.

In the preliminary study of the conditions upon which the results described in this paper depend, the ordinary grades of babbitt were found much too soft and yielding, under the necessary conditions of speed and oil feed, for such loads as 100 or 150 pounds per square inch of bearing surface. A very hard alloy was necessary, and one composed approximately of tin 90 per cent., copper 2 per cent. and antimony 8 per cent. gave satisfactory results. Then to maintain such conditions of load and speed that any oil could be broken down at any moment, it was found necessary in order to maintain a high degree of permanent evenness to repeatedly machine the journal and bearing to mechanically true surfaces. If such precautions be essential in precise quantitative observations, similar precautions are evidently necessary in factory operations.

Before the introduction of petroleum products the manufacturer was little concerned with the viscosity and other physical constants of lubricants, for with simple oils or greases of definite composition he knew their respective lubricating qual-

ities. Then imperfect lubrication was of less consequence, for the prices of products were higher and competition less severe than now. Then the principles of friction and the importance of its control were imperfectly understood. In manufacturing operations, transmission or transportation, modern high speeds and excessively heavy loads had not to be considered.

Heavy Lubricants.—A new era in lubrication began with the discovery that heavy hydrocarbon oils possessed viscosity, durability and stability under varying conditons of speed and load. The lack of knowledge of the hydrocarbons of which the lubricating oils were composed, and arbitrary methods of refining, delayed the production of suitable lubricants. Even now no fundamental changes have been made other than the introduction of heavier hydrocarbons from crude-oil territory recently developed. Pennsylvania crude oils, of the general composition C_nH_{2n+2} have always yielded excellent light spindle-oils mostly the hydrocarbons C_nH_{2n} and C_nH_{2n-2}. But such oils have not the body requisite for lubricants for heavy loads and cylinder friction. This need led to the compounding of petroleum products with rape or castor oils and animal oils or greases, and manufacturers grew to believe that no other products would answer. But the heavy lubricants, carefully distilled and treated, derived from Texas and California petroleum, are well adapted and are widely used for bearings and cylinders. It has been realized that the ideal would be a solid lubricant embodying the desirable qualities of the liquid products with a superior wearing quality, and a low coefficient of friction, and readily convertible to a form conveniently applicable to the various journals and bearings. Soapstone, asbestos, natural graphite, etc., do not entirely possess these fundamental properties. Greases compounded with graphite are useful on low-speed bearings and under heavy work. On cast-iron bearings natural graphite is excellent, acting as a surface evener of the porous metal; but on finer surfaces care is necessary that it should not clot and scratch or abrade the journal or bearing. Graphite has the desirable

unctuous quality and great durability, when very pure and finely divided. Whether, in the deflocculated form, the ultimate molecules or atoms have a freedom of movement analogous to that of liquid molecules under stress of friction, or whatever may be the reason, it is highly unctuous.

Graphite.—Dr. E. G. Acheson has produced an ideal solid lubricant in his deflocculated graphite, which is more unctuous than ordinary graphite and remains suspended in water and oils under certain conditions. Owing to its power of forming coherent films on journals, its wearing qualities and its ease of application, it is an efficient lubricant.

If a substance which contains non-volatile carbon is subjected to the high temperature of the electric furnace, all the other elements are readily volatilized and only Acheson graphite remains. The unctuous graphite, which is formed in the production of carborundum, is leafy in structure, coherent, and greasy or unctuous to the touch, is segrated and not readily disintegrated. Deflocculated graphite, obtained from coal or coke alone, is also highly unctuous, but very pulverulent and capable of extreme subdivision. It is readily deflocculated and when mixed with water is commercially known as 'aquadag,' when mixed with oil as 'oildag' (aqua- or oil-Deflocculated Acheson Graphite).

Under proper conditions deflocculated graphite remains suspended in water, and owing to its extreme fineness a very small amount so suspended, not over 0.35 per cent., serves as an efficient lubricant. It is remarkable that such a small quantity is readily distributed by the water between a journal and bearing, under a load of 70 pounds per square inch of bearing surface, and that it maintains an extremely low coefficient of friction under high-speed conditions.

Bearing Materials.—The metals composing journal and bearing have considerable influence on the question of lubrication even with the best lubricating film intervening. Cast-iron, steel, bronze and babbitt alloys are generally used. Extreme care is necessary with cast-iron bearings in high-speed work. In the accurate adjustment necessary in machine test-

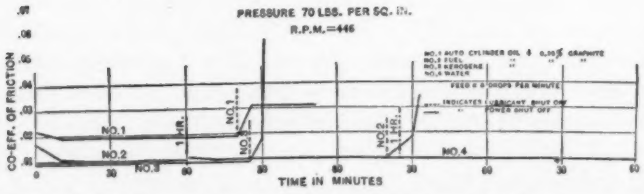
ing of lubricants, it has proved impossible to prevent injury to the journal when using a cast-iron bearing, nor is bronze altogether satisfactory. Properly-selected babbitt on a steel journal seems the most satisfactory. As stated before, for satisfactory lubrication both journal and bearing must be machined to true surfaces, kept smooth, scratches worked out and bare spots avoided. No factory supervision affords more desirable returns than constant, skilled attention to the condition of journals and bearings. Lubrication is the reduction of friction to the lowest percentage of the power used. A lubricant is an unctuous body that forms a continuous, coherent, durable film capable of holding apart rolling or sliding surfaces, and interposing the least possible resistance itself. The economic problem depends on the use of such lubricant under suitable conditions.

Lubricants Classified.—Commercial lubricants are water, oils, greases and solids. Oils include light spindle, heavy engine and cylinder products—either unmixed petroleum hydrocarbons, or mixtures of these with some one of the vegetable or animal oils, or with tallow or grease. The greases are generally classified according to their consistency, dependent on the proportion of lime or soda soaps mixed with a hydrocarbon oil as a carrier. Water has no oiliness, but under certain conditions is found to make the surfaces of cylinders extremely smooth and thus reduces friction. A practical knowledge of hydrocarbon lubricants should include both a knowledge of the crude oil from which it is prepared and the methods of refining. The best products are obtained by processes that leave the hydrocarbons essentially of the same composition that they had in the crude oils, but when the distillation causes decomposition the products are inferior. Improper refining can be detected by simple tests for the presence of free acid or free alkali, sodium sulphate or sodium salts of organic acids—all of which may be injurious to metallic surfaces. One of the greatest difficulties in refining is to treat with just sufficient caustic soda to remove acids and not to introduce an

excess which will form an emulsion. The best refined oils are those that contain a minute trace of alkali.

Lubricant Tests.—The ordinary tests for lubricating oils are the determination of viscosity, specific gravity, the flash and fire temperatures. There is no accurate method of determining the oiliness or greasiness by analysis, yet it is readily distinguishable and is a factor in the efficiency of lubricants. Authorities differ as to the most efficient methods of testing lubricating oils. Redwood thinks viscosity is the best guide to lubricating value, on account of the close relationship between viscosity and the laws of friction of liquids. As he failed to obtain satisfactory results with any friction-testing machine at his disposal, he concluded that their indications are misleading. He cited the lack of agreement between the work

FIG. 2.



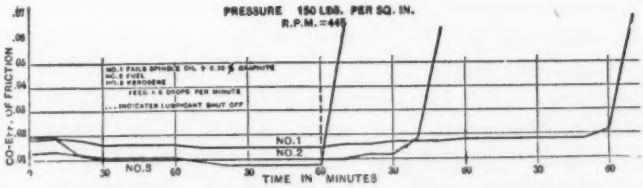
Curves of friction with varying viscosity of lubricant—bearing pressure, 70 pounds per square inch.

of Ordway and Woodward, with an apparatus constructed to apply pressures of 40 pounds per square inch, and that of Tower carried out under pressures of 100 to 600 pounds per square inch in an oil-bath system of lubrication, and the opinions of others on these results, to show that the conditions did not approximate those of actual practice. Hence he relies upon viscosity as the most valuable test of lubricating quality at our disposal. The value of viscosity is recognized, but all do not agree as to its practical reliability. Archbutt considers oiliness or greasiness nearly as important. He states that at very low speeds the friction of a journal should be proportional to the viscosity of the oil, but at high speeds, and therefore increased temperature, the relation of friction to speed

ceases and the viscosity is diminished with a proportional change in the carrying power of the journal. He holds that oiliness is of especial importance under heavy loads and high speeds. Hurst maintains that the test of an oil from a journal under the practical conditions of its use, shows conclusively its adaptability to such use.

The Carpenter Lubricant Tester.—The principal observations in mechanical tests are the effects of speed, load, temperature and the frictional effects due to viscosity and oiliness; the quality of the oil is measured by the frictional resistance, the temperature and the endurance of the oil film. The machine devised by Prof. Carpenter, used during the past year in these investigations, has fulfilled all requirements in tests of light and heavy oils, greases and graphite, owing to its

FIG. 3.



Curves of friction with varying viscosity of lubricant—bearing pressure, 150 pounds per square inch.

sensitive adjustment, durable efficiency, and its wide range of possible tests. The machine has an accurate adjustment for recording the speed, and a long lever-arm with a Vernier attachment graduated to tenths of a pound, for recording the friction. The load is applied by a powerful spring, worked by a cam and lever, up to 6,000 pounds total load.

The projected area of the bearing used is approximately 8 square inches; the journal is about 3 inches in circumference and nearly 1 foot long. A cast-iron frame babbitted and machined to a true surface was used. Even after careful machining continuous frictional work was necessary on the babbitt surface to reach the proper conditions of constant results. When satisfactory results were once obtained there was little

difficulty in keeping the surfaces of the hard form of babbitt in suitable condition. A thermometer was inserted in a hole in the bearing which extended close to the journal. Tests made at steam temperature, 210 degrees F. (*sic*), were carried out in a hollow cast-iron babbitted bearing, with steam attachments for maintaining the desired temperature. The lubricant was run in from a sight-feed cup through a small hole close to one side of the bearing, with careful regulation of the flow for proper adjustment of the feed. Two channels or grooves are run diagonally across the babbitt face from the inlet hole to allow of equal and even distribution of the lubricant; these channels must be carefully gauged for an even flow, or dry spots or streaks appear on the journal accompanied by a sudden greatly-increased friction as indicated on the friction bar. This detail requires careful and constant attention, for on it depends the continuous regularity of the friction curve. The journal is partially exposed to allow the observation of the formation of the film, its comparative thickness and any irregularity due to an imperfect journal or bearing, or improper lubrication.

The Case School Tests.—Accurate testing of the mechanical efficiency of oils presented an extensive field of labor, as there are no general standards of comparison under any conditions of operation. Such constants must necessarily be based on arbitrary data; nevertheless if they are accurately determined on a standard machine, with the conditions of the journal and bearing selected, the load and speed constants on this machine may be readily ascertained on any other equally efficient machine. These tests must be performed with a scientific accuracy of exact quantitative observations and with close supervision of all details. The conditions should be as close as is practicable to factory conditions.

The results obtained on water, kerosene and fuel oil as vehicles of graphite present novel and interesting features. Water alone serves as a lubricating film in steam cylinders, but since it is of no value on journals, the lubricating value of aqueous suspended graphite must be wholly due to the graph-

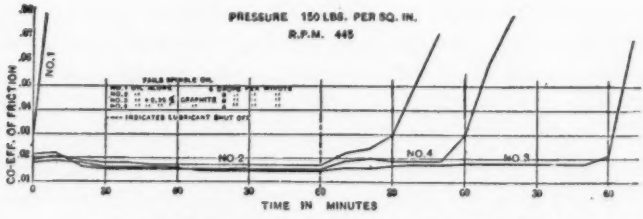
ite. The same is true of kerosene and fuel oils which alone are practically devoid of lubricating quality.

To test the effect of varying viscosity in lubricants and at the same time the lubricating quality of deflocculated graphite, tests were made with water, kerosene, a fuel oil and an auto cylinder oil, each carrying 0.35 per cent. of graphite. The results are shown by the curves in Fig. 2, in which the speed is maintained at 446 r.p.m. and the load at 70 pounds per square inch. Frictional load and temperature were observed at ten-minute intervals. On the chart (Fig. 2) the time is given in half-hour limits and the coefficient of friction in hundredths of a unit. The curve for water and graphite is practically a straight line, with scarcely any variations of the 4 hours shown. The test continued 15 hours with the same result. There were several stops, indicated by a dotted line on the chart; but there was no change in the direction of the curve by stopping and starting. Curve No. 3 for kerosense oil with graphite is also a straight line showing a very slightly lower coefficient than water. The curve for fuel oil and graphite is also practically a straight line and with an endurance test for 1½ hours after the oil supply was shut off; here the frictional coefficient is slightly higher than that of either water or kerosene. A similar regularity appears in the curve of the auto cylinder oil with graphite, but the frictional coefficient is very materially higher than those of the other lubricants, which may be considered as a measure of the comparative greater internal viscosity of the auto oil.

The effect of varying viscosity in lubricants and the lubricating quality of graphite under practically the same speed, 445 r.p.m., but with a load of 150 pounds per square inch, using kerosene, a fuel oil and a spindle oil with the same proportion of graphite and the same oil supply, is shown in Fig. 3. Kerosene here shows a very slight irregularity in its coefficient, differing slightly from that in the preceding chart. Here the greater internal viscosity of fuel oil is shown by the increased friction. No doubt the fuel oil possesses a slight oiliness, enabling it at the beginning of the test to indicate a

lower coefficient than kerosene. Kerosene maintains a higher coefficient for a few minutes until the continuous film of graphite has formed and reduced the coefficient to its normal condition. The spindle oil also possesses a certain oiliness, which enables it to begin the test with a coefficient that changes only slightly during the entire period, with an endurance test extending during 2 hours before the oil breaks and with only a slight increase in the coefficient of friction after the oil supply was shut off. The comparative endurance of the three oils is also worthy of note. While under a bearing load of 150 pounds per square inch kerosense maintains an extremely low coefficient, yet the fact that it breaks immediately when the oil supply is shut off indicates that it has not the power to form a coherent graphite film, which the fuel oil has to some extent and the spindle oil has in a marked degree.

FIG. 4.



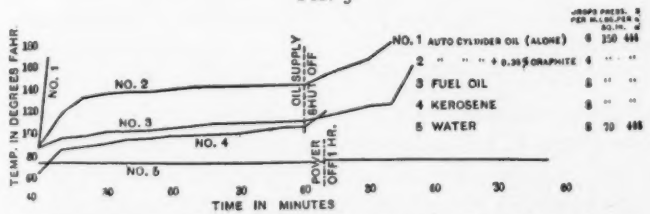
Curves of friction of oil and oildag with varying feeds.

Fig. 4 is a chart of tests runs at a load of 150 pounds per square inch and a speed of 445 r.p.m. It shows the effect of a variable feed when using spindle oil. In one test on the oil alone the supply was regulated to break the oil at the beginning of the test and also to perform its function as a lubricant. The effect of graphite on the lubricating quality of the oil is also shown in curves No. 3 and No. 4, No. 3 representing a feed of 8 drops per minute and No. 4 a feed of 4 drops per minute. The diminished coefficient in curve No. 4 as compared with No. 2 represents the lubricating effect of graphite, and this effect is still further shown by the increased endurance test in curve No. 4. It will also be observed that be-

sides diminishing friction, No. 4 is based on an oil supply due to the graphite of only one-half that of curve No. 2 of the oil alone.

In Fig. 5 the curves of temperatures are recorded in the friction tests presented in Figs. 2 and 3. The load is 150 pounds per square inch for auto oil, fuel oil and kerosene, and 70 pounds per square inch for water. Speed 444 r.p.m. for all except water, where it was 446 r.p.m. In the case of the auto oil alone there was an immediate rise in temperature corresponding to the breaking point of the oil as shown in the friction test. It is interesting to compare this temperature with that of curve No. 2 (auto oil with 0.35 per cent. of graphite) in which the temperature rises within 20 minutes to

FIG. 5



Temperature curves for lubricants of varying viscosity with or without graphite.

a definite point and then continues in a nearly straight line with little variation to the point where the oil supply was shut off at the end of 2 hours. Curve No. 3 of the temperatures of fuel oil and graphite also shows a very slight variation after 30 minutes, when the stable conditions of lubrication were established. A difference of approximately 25 degrees F. is shown between the curves of the auto and fuel oils, which must represent the larger escape of energy in the form of heat from the bearing due to the greater internal resistance of the auto oil. The temperatures of kerosene with graphite, shown in Curve No. 4, are approximately 10 degrees lower than those in the fuel-oil curve, due to the still smaller resistance of the kerosene. Considering the small difference between the specific gravity of fuel oil (about 35 degrees Baumé) and that of

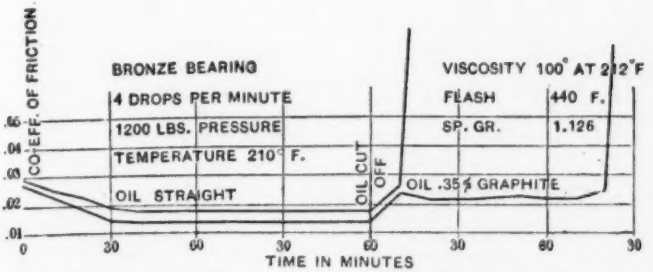
kerosene (about 45 degrees Baumé) the difference in temperature of these two curves is a good example of the accuracy of observation possible in these tests. Perhaps the curve of temperatures for water and graphite is the most striking on this chart; this only shows 4 hours, but actually lasted 15 hours with several stops, but the temperature at the start was the same as that at the time of interruption. This chart shows an extremely low temperature, 65 degrees F., practically room temperature, which is never exceeded by more than 5 degrees F. and is essentially a straight line from start to finish. The use of water as a vehicle for graphite does not interfere with the best work that graphite is capable of performing.

Considerable attention was given to the behavior of heavy engine and cylinder oils, both plain hydrocarbon and compounded oils. A special form of bearing was constructed—a cast-iron frame with a hollow chamber for introducing steam, and a face of hard babbitt. For some tests a bearing of similar construction but with a bronze face was used. The results were generally less satisfactory with the bronze than with the babbitt bearings, not only with heavy oils, but with others. Hard babbitt is better than bronze. Three cylinder oils, American, Galena and "600 W.," alone and with graphite, were tested for their frictional coefficients and endurance, in a continuous run of 2 hours. The physical constants, especially specific gravity and viscosity, of these oils are given for comparison.

Fig. 6 is a chart of a straight hydrocarbon oil (American cylinder oil) with the bronze bearing; the lubricant was supplied at the rate of 4 drops per minute, the total pressure was 1,200 pounds and the speed 245 r.p.m. For the oil alone the curve begins at a somewhat higher coefficient than is maintained after the first half hour and then proceeds in a straight line without variation till the feed is stopped. The endurance run was doubtless shorter than it would have been with babbitt bearings, as was demonstrated by test. The curve with oil and graphite follows this very closely, but with a considerable diminution of the coefficient of friction.

Fig. 7 is a diagram of the tests with 600 W. cylinder oil alone and with graphite. This oil has a lower specific gravity and somewhat higher viscosity than the last one. The tests were run at the same total pressure and the same speed as the previous ones, but with double oil feed and with babbitt bear-

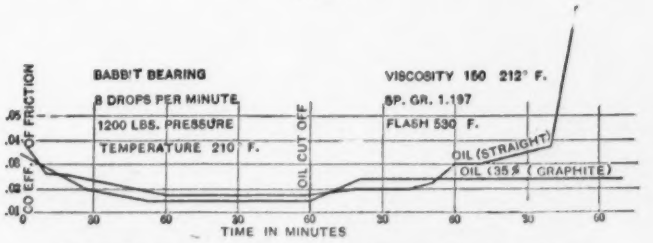
FIG. 6.



Curves for American cylinder oil with and without graphite.

ings. Owing to its greater viscosity the straight oil showed a considerably higher coefficient at the start and the test ran for one hour before the oil reached normal conditions, which it maintained until the feed was stopped. After the oil was shut off the lubrication was maintained with irregularity and

FIG. 7.



Curves for "600 W." cylinder oil with or without graphite.

increased friction during 1 hour and 40 minutes, when it broke. The curve of this oil with 0.35 per cent. graphite is very similar, but it reaches normal conditions sooner, continues in a straight line till the supply is stopped, and then with slightly

increased friction continues in a straight line. The marked influence of graphite is plainly apparent in these curves.

Galena cylinder oil was tested with and without graphite under the same conditions. This is less viscous but heavier

FIG. 8.

Curves for Galena cylinder oil with or without graphite.

than the preceding oil. To reach normal conditions the straight oil required 1 hour, and oil with 0.35 per cent. graphite, 45 minutes. After feed was stopped the oil alone broke in 1½ hours, the oil with graphite broke after 4½ hours.

Conclusions.—The results of these tests with regard to the use of graphite as a solid lubricant indicate that it can be applied with advantage in the deflocculated form. Its most characteristic effect is that of a surface evener, equalizing the metallic depressions and projections on the surfaces of bearing and journal, and endowed with a certain freedom of motion under pressure. The great efficiency of graphite in automobile lubrication, by increasing engine power and controlling temperature and wear and tear of bearings, has been demonstrated in a series of tests conducted by the Automobile Club of America. In connection with the reduction in friction the extremely small proportion of graphite necessary is notable, viz: 1 cubic inch of graphite in 3 gallons of oil.

In these observations and in all the work that has been done in this field, there is not a more impressive example of the efficiency of graphite in lubrication than that presented in the curves of friction and temperature of water and graphite, for the water serves merely as a vehicle and is completely devoid of lubricating quality; the graphite performs its work without aid and with no limiting conditions.

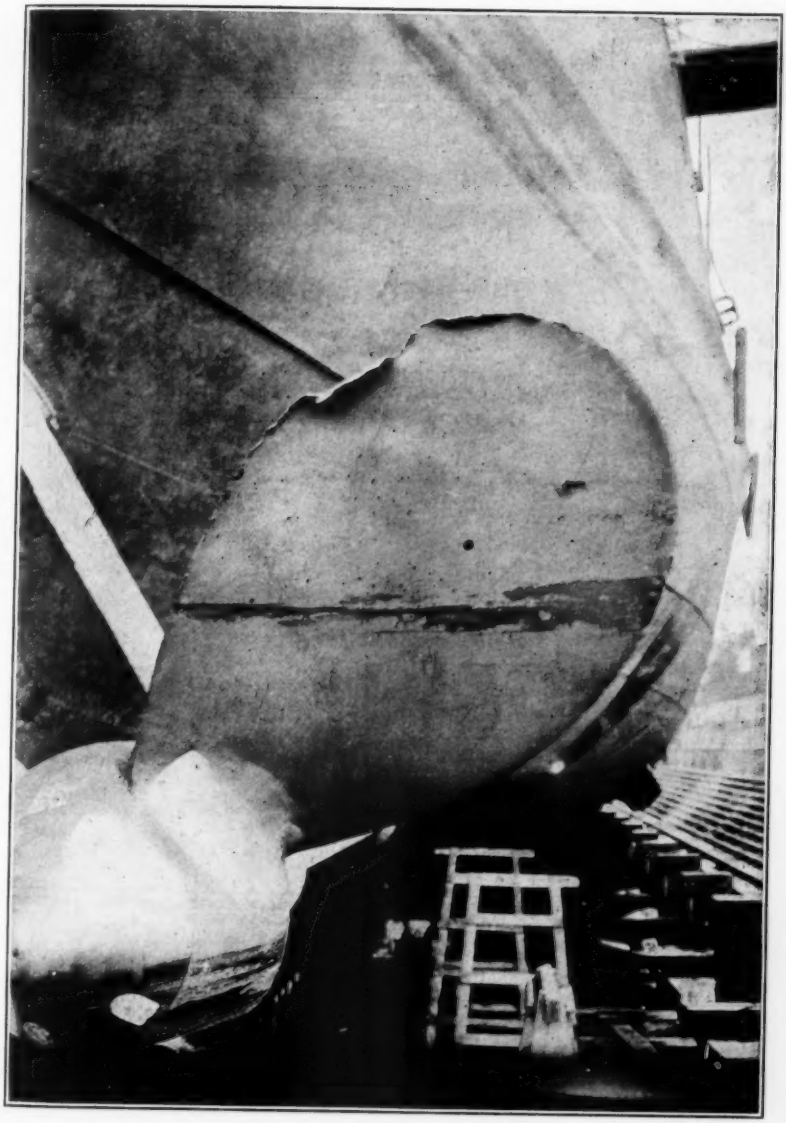


Fig. 1.

WORST BLADE OF STARBOARD PROPELLER.

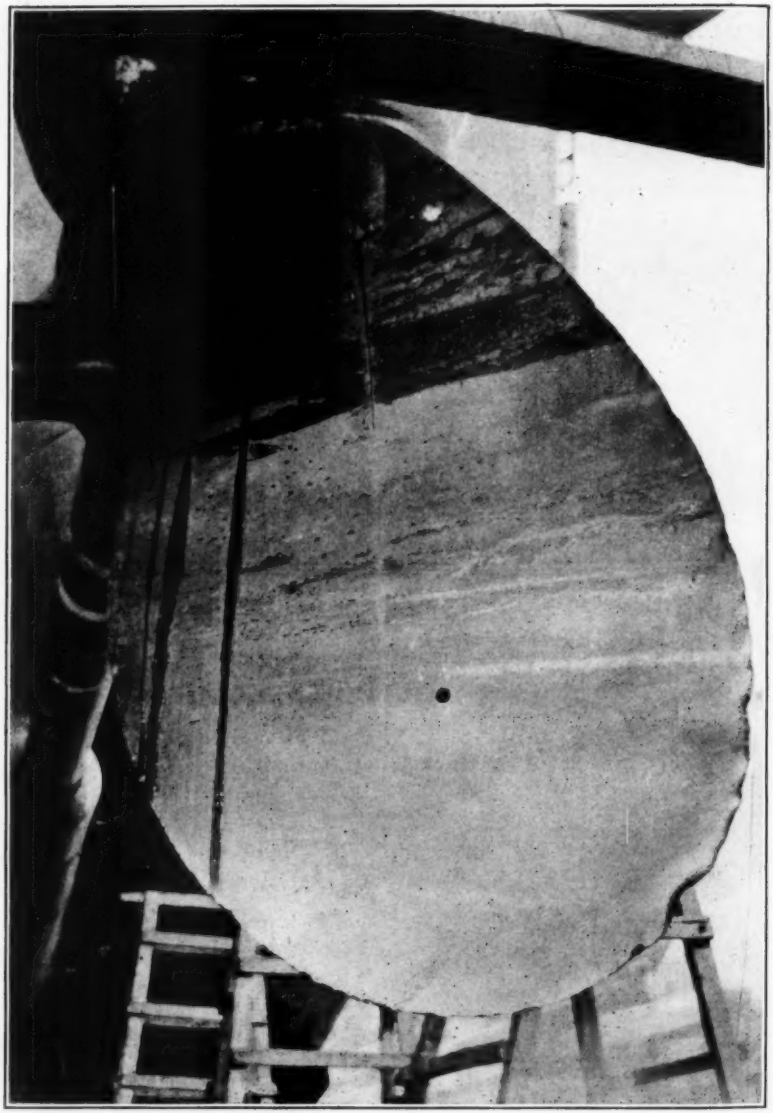


Fig. 2.

SECOND BLADE OF STARBOARD PROPELLER.

The third blade was the best of the three and was about in the same condition as the port blades, viz: slightly crimped on the edges, but not deformed in any other particular.

EFFECTS OF INJURY TO PROPELLERS U. S. S. "MICHIGAN."

While running in for anchoring during preliminary trials and after standardization trials had been run, the U. S. S. *Michigan* ran aground in a fog near Provincetown, Mass., injuring the blades of starboard propeller more than those of the port. The photographs show the extent of damage to starboard blades.

After injury an attempt was made to run the 24-hour trial at 12 knots, when it was found that the power required to make the revolutions which the standardization curve showed to be necessary to give 12 knots was about 35 per cent. more than the power required from standardization. The following data of power of engines under the different conditions is given:

STANDARDIZATION DATA, JUNE 9, 1909.

Average revolutions, both engines	73	
Speed, knots	12.15	(from curve.)
I.H.P., starboard	1,666	
port	1,794	
total	3,460	

12-KNOT RUN.

Revolutions	73	
Speed, knots	...	
I.H.P., starboard	2,698½	
port	2,052½	
total	4,751	diff., 646.0
	3,460	
Increase	1,291	
Increase, per cent	37.3	

NAVAL NOTES.

ENGLAND.

New British Battleships.

In discussing this subject "Engineering" says:

The first of the desiderata for battleships laid down by the strategist is gun power. In his paper at the Institution of Naval Architects, Admiral Bacon seemed to regard this as the dominant consideration, and was prepared to sacrifice even armor protection to a greater or less extent. There are some indications in our later ships of the acceptance of this view, although the more vital parts of the ship in the *Orion* class are still protected by 11-inch plating. It is accepted that with the modern 12-inch gun no armor of reasonable thickness can be effective in keeping out armor-piercing projectiles at 6,000 yards or less, and that when fighting at any range beyond this it will not be possible to achieve decisive results. The extent and effect of damage done to the armor of different thicknesses at such relatively close range will require very serious consideration, and much courage will be involved in the acceptance of any diminution in thickness of armor for the protection of the fighting mechanism and for the maintenance of a due reserve of buoyancy. That being so, we do not anticipate any check on increasing displacement due to a less percentage of the tonnage being allotted for protection. Admiral Bacon was probably quite right in his view that minute subdivision would give the larger ship a better chance of withstanding attack by the torpedo than the smaller ship, and from this point of view the latter ships are at least as satisfactory as the former vessels. There is, however, no tendency at present to utilize light armor as a protection under water against torpedo attack, minute subdivision being preferred.

Like the *St. Vincents* and the *Neptunes*, the later ships

have only ten guns in their primary armament, and these will be, at all events in the great majority of cases, of 12-inch bore and 50 calibers length; while the torpedo-repelling guns will be of 4-inch bore. In the choice of guns the Admiralty are influenced by the gun-control system, and are not swayed in any way by the tendency of other Powers to adopt a larger gun in addition to, or in supplement of, the 4-inch breech-loader. In their earlier *Dreadnoughts* the Germans followed the practice adopted in the *King Edward* of fitting as secondary armament guns corresponding to our 6-inch quick-firers. The United States authorities seem to have a preference for a 5-inch gun, and in France the 5½-inch is spoken of as part of the armament of the new ships, while Japan is adopting the 6-inch gun. It has not been made clear, however, whether these several navies are fitting the larger weapons as part of the armament for use in line of battle, or exclusively—as in the case of the British 4-inch breech-loaders—for repelling torpedo attack. Certainly, the 6-inch guns would be used against a battleship if favorable conditions arose, but the contention is that when such opportunity arises, the 12-inch guns would be more effectively used without the 6-inch guns being brought into service, than with the latter in use.

What the modern tactician calls for is not so much penetrating power as great bursting charge, and in this respect the damage done by one well-placed shot from a 12-inch gun would be superior to any number of 6-inch shots which could be driven home in the same period of time. Moreover, it is contended that the confusion due to the flight of shot of various sizes used simultaneously seriously militates against effective "spotting" and gun-control, and, therefore, against accuracy. In short, it is accepted from the evidences of recent battle practice that, because of this more effective gun-control, a *Dreadnought* can, with her 12-inch guns, secure a greater number of hits in a given time than a *King Edward*, where the number of guns is greater and their possible rapidity of fire higher, because the indiscriminate flight of 12-inch, 9.2-inch and 6-inch projectiles is confusing to the "spotter"

in the gun-control station. This is a point upon which the Admiralty probably have more experience than anyone, not even excluding the authorities in foreign navy departments; and as our naval tacticians have continued, in the ten recent ships laid down for the Imperial and Colonial Navies, to fit one caliber of large guns in association only with 4-inch torpedo-repelling guns, there is more than a presumption that this arrangement is likely to give the most effective results.

Perhaps the most important departure made in the armament of the new ships is in connection with the disposition of the primary guns, and this raises the question of the importance of bow fire. This is one of those many questions of naval constructional policy which have recurred at intervals ever since gunships were built. When 6-inch guns were introduced, one was placed on each quarter and several on the broadside, and in some cases these were in two-storied casemates, as in the *Powerful* and *Terrible*. With the introduction of the 9.2-inch gun in supplement of the 12-inch weapon, as in the *King Edward* class, the smaller of the primary guns was similarly mounted on each quarter, often with 6-inch guns at a lower level. This practice was carried out also in the *Lord Nelson* class, where two 9.2-inch guns were mounted on each quarter, giving as bow fire two 12-inch and four 9.2-inch guns. The same principle has been adopted by the German naval authorities in their *Dreadnought* battleships and *Dreadnought* cruisers. The disadvantage of this, in their case, is that of the twelve 12-inch guns mounted only eight can be fired on the broadside, although six may be fired ahead or astern. The greatest development in bow fire, however, is in the case of the Brazilian battleships, where eight 12-inch guns may fire ahead. There is a pair on each broadside, and two pairs in the center line forward and two pairs aft, one pair in each case being mounted at a higher level, in order to fire over the pair of guns in front. The disposition of the large guns in the later British battleships greatly reduces the bow fire, the presumption being that a heavy broadside is regarded as more important.

In the earlier ships of the *Dreadnought* class the placing of the whole of the 12-inch guns on the middle line was considered—a practice which has since been adopted with the latest ships in the United States Navy. In this case also some of the guns were at different levels, so that four might fire ahead and four astern, and all ten guns on either broadside. In the British service, however, it was considered desirable to have a turret on each broadside, and thus, in the *Dreadnoughts* now completed, the wing guns fire ahead or astern, and on the broadside. In this way eight of the ten guns are utilizable on either broadside in an engagement in line formation. This arrangement had considerable advantage in the planning of the interior of the ship and the disposition of magazines. In the cruisers of the *Inflexible* class, however, where considerations of weight and speed necessitated a reduction of the total number of guns to eight, it was decided to adopt for the amidship guns the echelon emplacement, so that the guns in the waist of the ship could fire on either broadside; all eight guns are thus utilizable, while six fire ahead and six astern. The arc of training, however, of the amidship guns was unavoidably limited; and in the later battleships all ten guns are placed in the center line. This has necessitated a considerable increase in the length of the ship, but it simplifies the location of the torpedo-repelling guns, and may result further in affording them a larger measure of armor protection than is possible when the majority of them have to be distributed over the upper works. There is a difference of opinion as to whether these guns should be protected, but in view of the experience at Tsu-Shima, when a torpedo raid was made after the main engagement of the day, an effort should certainly be put forth to so protect these guns that they would be available after the main battle, in order to meet such a subsequent torpedo raid as is certain whether or not the main engagement has been decisive. It has been suggested that these lighter guns might be fitted with disappearing carriages, so that they could be carefully housed during the main action; but, so far, little seems to have been done in the application of the idea.

We have said that the length of the battleships has been increased. The *Formidables* were only 400 feet long, the *King Edwards* were increased to 425 feet, the *Dreadnoughts* to 490 feet, and now the later ships are 545 feet in length. The beam has not been increased to the same relative extent, as now the proportion of length to beam is 6.15 : 1, as compared with 5.35 : 1 in the case of the *Formidable*. One advantage of the longer ship is that a higher speed is achieved for a given power, while the addition to beam enables the weights to be carried at a higher level without affecting stability. The draught has remained practically constant, being 27 feet 6 inches. This, of course, is almost an imperative condition, since the vessels may have to fight in water where a greater depth would be a serious disadvantage.

Notwithstanding the increased displacement, the Admiralty have very properly determined that the speed should not be diminished; and thus the new ships, like those which have preceded them, will have a legend speed of 21 knots. Experience suggests that the actual speed will be 22 knots, since all the *Dreadnought* battleships tried have maintained practically this rate throughout their eight hours' trial. So far as the machinery is concerned the developments have been largely in detail. The Parsons turbine is being adopted for all of the ten capital ships recently ordered, with four shafts, each shaft having one ahead and one astern turbine. The astern turbine on the shaft having the high-pressure ahead turbine is separate, but the low-pressure ahead and astern turbines are incorporated in the same casing. The two ahead and two astern turbines on each side of the center line work in series both for ahead and astern driving. In order to secure the greater range of expansion necessary at low powers the high-pressure ahead turbine has been increased in length, and cruising turbines dispensed with. At higher speeds the steam is by-passed to the intermediate stage in the high-pressure turbine, but at low powers it is admitted at the high-pressure end. This practice has proved most satisfactory in electric-power installations, and is sure to give corresponding satisfaction in

warships. It has the advantage of simplifying the engine-room arrangements, and weighs slightly less. The adoption of the astern turbines on all four shafts greatly simplifies maneuvering, in addition to giving greater power, while all four propellers, instead of the wing propellers only, are used for going astern. Babcock & Wilcox boilers are used in some of the large ships, and Yarrow boilers in others, and it is satisfactory to note that the high speed maintained by those battleships is not achieved by overpressing the boilers, which, however, show a tendency to be made smaller. In the case of the Babcock boilers, the heating surface allowed per unit of power is 2¼ square feet, and in the Yarrow boilers 2½ square feet, the ratio of heating surface to grate area being in the former case 35: 1, and in the latter 60: 1, while something like 13 shaft horsepower is developed per ton of machinery, which is most satisfactory.

It will thus be seen that the new ships promise to be most efficient in every respect. No doubt differences of opinion, as has already been indicated, exist regarding the advantage possible with the higher rate of fire of smaller guns, introduced in some navies as secondary armament; but as the question has had careful consideration in the light of the fullest knowledge, the policy of the tacticians at the Admiralty must be accepted. At the same time the armor protection has been maintained. The conditions evolved by the tacticians have, moreover, been embodied in the ships by Sir Philip Watts, K.C.B., the Director of Naval Construction, and in the machinery by Engineer Vice-Admiral Oram, C.B., the Engineer-in-Chief of the Fleet, in such a way as to ensure the highest efficiency both from the stability and propulsion standpoints.

FRANCE.

The Projected French 23,400-ton Battleships.

After prolonged consideration and in spite of many contradictory rumors, the Consiel Supérieur of the French Marine has decided that the 23,400-ton battleships, which are to

be laid down during this year, shall have a primary armament of twelve 30-cm. guns (say, 11.8-inch), in six turrets, arranged two turrets forward, two aft—one of these in each case being able to fire over the other—and two amidships, one at each side. As secondary armament, there are to be eighteen 14-cm. (5.6-inch) guns, arranged in groups of three in armored casemates—one group on each side forward, one group on each side amidships, and one group on each side aft.

It was originally intended that these new battleships should have a displacement of 22,400 tons, but the number of the guns in the secondary armament has been raised and the armor-belt increased. The latter extends now from end to end of the vessels, and has a maximum thickness amidships of 27-cm. (10.63 inches) and a minimum thickness fore and aft of 18-cm. (7.08 inches). It rises 2.35 m. above the water line, and is taken down to 2.50 m. below it. The upper armored deck above the thick armor belt varies in thickness up to 5-cm., while the lower armored deck has a thickness of 7-cm, where it joins the lower edge of the main belt. The casemates of the 14-cm. guns are protected from top to bottom with 18-cm. armor, which also affords a considerable amount of protection to the funnels. Below the main belt armoring of the same thickness as that of the belt is arranged for about half the length of the vessel in order to protect the engines, boilers and the principal auxiliaries, dynamos, etc. Thick transverse bulkheads in front of the forward turrets and aft of the aft turrets complete the principal armoring. Lastly, the doubtful protection which it was sought to obtain in the *Danton* class by means of interior armoring, so as to diminish the effect of torpedoes, has been suppressed; but, on the other hand, considerable care has been taken to make the cellular subdivision of the triple bottom as perfect as possible, and it would appear to be probable that with a less weight the security obtained will not be less.

The various increases in guns and armor to which we have drawn attention account for the 1,000 extra tonnage in the displacement as compared with that of the ships as originally

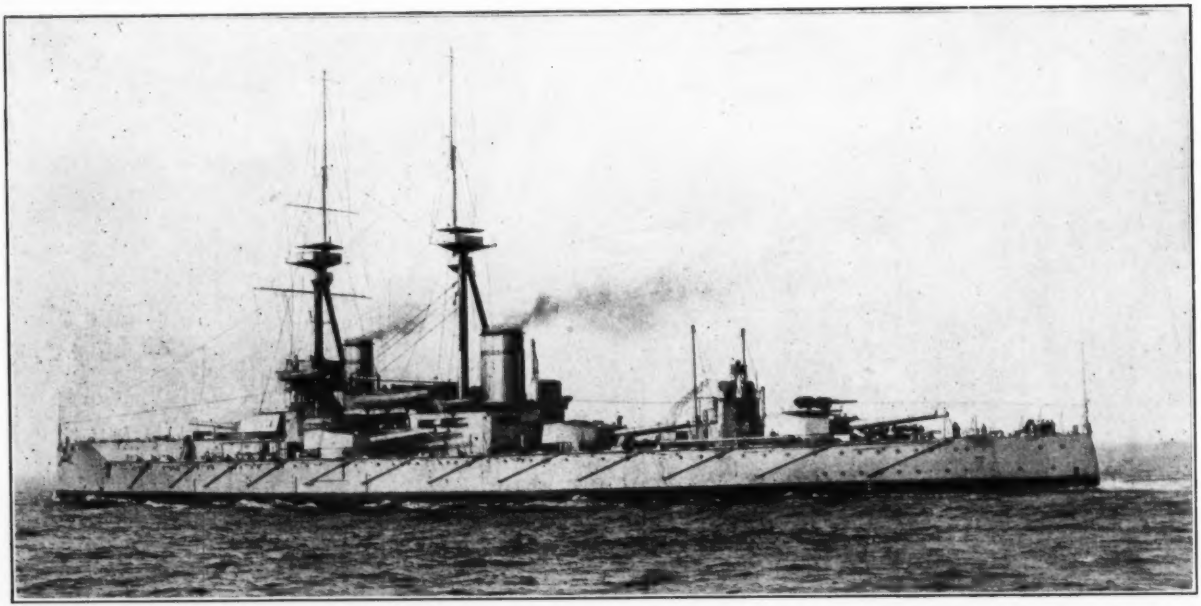
contemplated. Our esteemed contemporary, "Le Yacht," in commenting upon these vessels, says that this increase is by no means negligible even from the point of view of cost, as it means an extra 3 millions of francs for each vessel; but, it continues: "What should be considered is less the actual cost in money of the increase than its relative value as compared with that of the total displacement. Under these conditions it is difficult to say that the extra safety has been purchased at too great a cost. The higher the price of a warship becomes the greater are the sacrifices consented to in order to prolong to the utmost limit its offensive properties." This journal then goes on to remark that certain foreign writers have criticised these vessels before the actual details were properly known, and have asserted that certain recent battleships have attained an equal or even greater power with a less displacement. "It is possible," it continues, "that this criticism is in part well founded, though certainly to a less degree than is averred, for in France the displacement is always taken as referring to a completely armed vessel with its full load of coal." Abroad, on the other hand, as it points out, displacement is calculated with a sensibly less load of fuel, and it is hence impossible without accurate evidence to compare the differences which result from these two customs, the influence of which may be considerable when it means, as it does in France, that the weight of the coal carried may be as much as 2,700 tons. This, it claims, may well account for the difference between the tonnage of the new French battleships and that of the latest British vessels. Our contemporary must, however, have overlooked the fact that even though British vessels may not have on board at the time of their trials the full amount of coal they are capable of carrying, they must be loaded with other weights to bring them to the contract draught line.

On the other hand, the French ships with their twelve 30-cm. guns can only fire ten on the broadside. But all the guns have a wide arc of fire—namely, 135 degrees on each side for the axial guns and 180 degrees for the amidships weapons.

Guns placed on the center line amidships in a vessel can only with difficulty fire through an arc of 90 degrees. To get more than this would mean that the length of the ship must be increased, and, remarks our contemporary, we have already arrived at dimensions which are so formidable that we rightly hesitate to increase them. The latest American battleships, with six double turrets, are to have a displacement of 26,000 tons, and the arc of fire of these turrets is very limited. The *Neptune* and the *St. Vincent* can only fire eight guns on the broadside. It is claimed, therefore, that the French vessels appear to have their main armament very well arranged.

The employment of the 30-cm. gun on these vessels has met with adverse comment in some directions. It has been urged that larger weapons should be employed, and attention is drawn to the fact that Great Britain proposes the use of 34-cm. and the United States of 35.5-cm. guns. The reply of "Le Yacht" to this comment is that nothing is known as to how these bigger guns will behave, and it would take a lengthy series of trials to find out. Further than this, that the 30-cm. gun is powerful enough to penetrate at the extreme ranges at which fighting is practicable any armor at present existing or to be employed in vessels now being constructed. On the other hand, it is pointed out that a bigger gun than the 14-cm. might well have been chosen for the secondary armament.

The new vessels are to be fitted with turbine engines, which are calculated to give a speed of 21 knots, or even more. The radius of action with the full complement of coal, etc., is to be 8,400 miles. The boilers are to be large-tube boilers of a French type, probably either Belleville or Niclausse, and there are to be four torpedo tubes for 450-mm. torpedoes. The crew will consist of 930 officers and men.—"The Engineer."



H. M. Battleship "Vanguard."

SHIPS.

ENGLAND.

The Vanguard is the fifth battleship of the *Dreadnought* class to be completed, and the view given is interesting, as it shows some of the new features which may be seen by any one now that the vessel is removed from the veil of secrecy which is supposed to surround all our naval construction works. Four pairs of 12-inch guns are visible: the two on the forecastle level forward, the two on the port quarter, which may fire ahead in line with the keel owing to the forecastle being cut away as shown, two in the center line a little abaft of midships, which have a wide arc of training, but cannot fire ahead or astern, and the two on the center line on the quarter deck. There are also two on the starboard quarter, so that eight may fire on the beam, six ahead and six astern. According to the Navy Estimates there are, in addition, twenty 4-inch breech-loaders, and several of these can be seen projecting through the gun ports of the citadel which surrounds the base of the funnels, where also the boats are located. Forward of this is the main navigating station, while at the aft end there is an additional conning tower. Forward of the aft gun house there is seen a steel structure which serves several purposes. On the top is the aft navigating station, with searchlight, and gun-control station, supplementary to those on the masts, while the interior is for hoists, etc. The arrangement of wires for aerial telegraph communication is also an interesting feature.

The dimensions of this ship, which has been built by the Vickers Company at their Naval Construction Works at Barrow-in-Furness, as given in the Navy Estimates, are: Length between perpendiculars, 500 feet; breadth, 84 feet; mean load draught, 27 feet; displacement, 19,250 tons; weight of hull,

including armor and backing, 11,950 tons; designed horsepower, 24,500; speed, 21 knots. On the official trials, however, the vessel attained a mean speed on the measured mile of within a fraction of 22 knots, the steam consumption of the turbines being 13½ pounds per horsepower per hour. These results are exceptionally favorable. What, however, is of most interest at the present time is the period occupied in the building of this latest of *Dreadnoughts.* From the date of laying the keel until the date of commissioning, a few days hence, only twenty-three months have elapsed. This is a shorter time than in the case of any of the *Dreadnoughts,* excepting only the prototype, for the machinery of which the Vickers Company were the contractors; it is admitted that the full resources of Portsmouth Dockyard were concentrated on the hull construction, whereas the Barrow establishment, during the period of construction of the *Vanguard,* has been employed on much equally important work, to be completed with corresponding expedition. Only exceptional resources and the avoidance of labor difficulties can ensure the completion of a *Dreadnought* within two years. Of this fact the appended table affords testimony. It shows the dates of the laying of the keel and of commissioning of six *Dreadnoughts,* and the time occupied in their construction.

Name.	Where built.	Keel laid.	Commissioned.	Time taken.
				Months.
Bellerophon.....	Portsmouth.	Dec. 3, 1906.	Feb. 20, 1909.	26½
Téméraire	Devonport.	Jan. 1, 1907.	May 15, 1909.	28½
Superb.....	Elswick.	Feb. 6, 1907.	May 29, 1909.	27½
St. Vincent.....	Portsmouth.	Dec. 30, 1907.	March, 1910.	27
Collingwood.....	Devonport.	Feb. 3, 1908.	March, 1910.	26
Vanguard.....	Barrow.	April 2, 1908.	Mar. 1, 1910.	23

The *Vanguard* has been completed just within the 23 months, but the other ships have taken from 26 to 28½ months. It is true that labor disputes have influenced the results, but these are contingencies which must be anticipated. In arranging programs to ensure our required standard of strength relative to any foreign fleet at any future date, 27

months would seem, therefore, to be the minimum period allowable, notwithstanding the creditable performance of the Vickers Company.—"Engineering."

The Tribal Class of Destroyers.

The *Zulu* is one of the third batch of the Tribal class of destroyers, of which a dozen have now been put afloat. These vessels of the 1907-8 program were not ordered until January, 1908, and they were not actually laid down until much later—the *Nubian* on May 18th, the *Viking* on June 11th, the *Crusader* on June 22d, the *Maori* on August 6th, and the *Zulu* on August 18th. The *Zulu* was held up for practically the whole of 1908 owing to the strike on the North-East Coast, and it was only on September 16th of last year that the vessel was launched. She has a length of 280 feet, a beam of 27 feet, a draught of 8 feet 10½ inches and a displacement of 1,030 tons. She has Parsons turbines of 15,500 shaft horsepower, built at the St. Peter's works of Hawthorn, Leslie & Co., Limited. There are three propellers. Steam is raised in Yarrow boilers burning oil fuel, of which 190 tons are carried. The armament comprises two 4-inch quick-firing guns and two 18-inch torpedo tubes. The vessel proceeded to sea on November 29th last for her first preliminary trial, when she attained the contract speed. Owing to adverse weather the official trial could not be carried out until December 28th, on which occasion a speed of 33½ knots was easily reached in deep water, while the fuel consumption was well within the specified limit. The cruising trial of twenty-four hours' duration began the next day, and was carried through satisfactorily, the radius of action being fully 20 per cent. over contract requirements.

Of the five ocean-going torpedo-boat destroyers provided by the 1905-6 program, two were built on the North-East Coast—the *Ghurka* by Hawthorn, Leslie & Co., Limited, and the *Afridi* by Sir W. G. Armstrong, Whitworth & Co., Limited. The completion of both these vessels was seriously de-

layed by the labor disputes, but the *Ghurka* was handed over to the Admiralty before the close of 1908, and was attached to the First Division of the Home Fleet as from March 1st of last year. In "The Engineer" of November 1st, 1907, was published an illustration and some particulars of the constructional features of the *Ghurka*, and to supplement these and also to afford a comparison with the later vessel of the Tribal class of destroyer the following additional data will be interesting: The *Ghurka* has a length of 255 feet, a beam of 25 feet 6 inches and a depth of 16 feet. The propellers are supported on brackets, the central propeller—which is driven by the high-pressure turbine—being at a lower level and abaft the side propellers, which are attached to the shaft, on which is mounted a cruising and a low-pressure ahead turbine and an astern turbine. The hull is constructed almost entirely of high-tensile plates and angles. The armament consists of three 12-pounder guns and two 18-inch torpedo tubes. There are three boiler rooms, containing five water-tube boilers of the latest improved Yarrow type, and on each side are oil-fuel tanks for use in peace time only. The boilers are worked on the closed-stokehold system with forced draft. There are two fans in each stokehold driven by forced-lubricated engines, whilst the oil fuel is supplied to the furnaces by an oil pump fitted in each stokehold. The oil-fuel tanks are placed at the forward and after ends of the machinery spaces.

The Parsons turbine of the *Ghurka* was built by Hawthorn, Leslie & Co., was made to duplicate gauges, so as to ensure interchangeability with the other four sister vessels of the Tribal class of destroyers. There are seven turbines, developing from 14,000 to 15,000 indicated horsepower when running at full speed, a high-pressure cruising, an intermediate cruising, a high-pressure main, and two low-pressure main turbines, all for going ahead, in addition to two astern turbines. One cruising, together with one low-pressure main ahead and one astern turbine, is mounted on each wing shaft, whilst the high-pressure main turbine is on the center shaft. At low power the sequence of steam is through the high-

pressure cruising to the intermediate-pressure cruising, thence to the high-pressure main, and to the low-pressure main turbines. For intermediate speeds the high-pressure turbine is cut out, the steam from the boiler entering the intermediate-pressure cruising turbine, thence going to the high-pressure main turbine and to the low-pressure main turbine. For full speed both cruising turbines are out of action.

The condensers are arranged in the wings of the ship and the circulating pumps at the forward end, the evaporating of the distilling plant being also arranged in the wing on the forward engine-room bulkhead room. It may be here recalled that the conditions laid down by the Admiralty in respect of these "ocean-going" destroyers required the maintenance for six hours of an average speed of 33 knots, and that on the 24 hours' consumption trials, the vessels should have a radius of action of 1,500 miles at a speed of not less than 13.5 knots per hour. The speed of 33 knots is an exceptionally high one under any conditions, but with the stipulation that this speed should be maintained over a period of six hours, combined with heavy scantlings of hull and machinery, and with strictly limited consumption of fuel—its performance becomes a matter of no little complexity.

On the official full-speed trial the *Ghurka* attained a mean speed of 33.997 knots at 772 revolutions, and at a satisfactory rate of consumption of oil and water. This speed could have been considerably exceeded but for the restriction as to the quantity of oil to be burnt per square foot of heating surface. The further trials for consumption at cruising speed, steering, firing, etc., as well as that for testing the behavior of the vessel in a seaway, were also carried out to the requirements of the Admiralty.

It will be seen that the *Zulu* is a substantially larger boat than the *Ghurka*. At the time the first "33-knotters" were built, the problem was to get 33 knots for a sea-going destroyer on the most moderate dimensions, and in this, Hawthorn, Leslie & Co., achieved some success, as the *Ghurka* is the smallest boat of the class and has only five boilers. Some

of the other vessels are very much larger and are fitted with six boilers. They of course attained somewhat higher speeds. In the *Zulu* there are six boilers, and the builders have gone to dimensions which will make the vessel capable of doing scout work.—"The Engineer."

JAPAN.

The Japanese Battleship Satsuma.

The Japanese battleship *Satsuma* is of special interest for a variety of reasons. Her conception dates from the outbreak of the Russo-Japanese War, and she altogether pre-dates the *Dreadnought* in the matter of being the first attempt to realize Colonel Cuniberti's "Ideal Battleship." The idea of the famous Italian naval architect first saw the light in July, 1903.

The Russo-Japanese War commenced in February, 1904. Its duration was then uncertain; the loss of Japanese battleships probable. It was, therefore, imperative that Japan should lay down warships, it was equally imperative—or at least obviously desirable—to lay down something altogether superior to any possible enemy. Colonel Cuniberti, in sketching his ideal, mentioned this ideal ship opposed to a fleet; Japan undoubtedly had this in mind when the *Satsuma* and *Aki* were projected.

It is a most uncertain point whether such ships would have been imagined in 1904, but for the war; it is practically certain that but for the war they would not have been laid down in Japan, but would have been built at Elswick or Barrow. Owing to the war and its uncertain duration, the ships had to be built in Japan, *nolens volens*—the second point of interest about the *Satsuma*.

Japanese naval architects, being without experience in designing large battleships, and the necessary material having to be collected, it was not till 1905 that the ships were actually laid down—the *Aki* in March; the *Satsuma* in May.

At about this time the issue of the war was no longer in

doubt. This, together with the reasons mentioned above, accounts for the long time that the *Satsuma* has been under construction. Her sister, the *Aki,* was further delayed by the decision to fit her with turbines.

It will be noted that the *Satsuma* is a development of the *Kashima* and *Katori,* which vessels closely resemble our own *King Edward* class, though, like all warships of Japanese construction, she has the extraordinary dipping schooner bow. The precise object of this schooner bow is not known, but possibly the idea is to give Japanese vessels a distinctly recognizable appearance. In the past, ships firing into friends by mistake has happened on more than one occasion, and is believed to have happened at least once in the late war. From this point of view characterization is valuable.

Like the *Kashima* and *Katori,* the *Satsuma* has only comparatively moderate armor protection. The belt remains at 9 inches, whereas we, in the *Lord Nelson,* had previously gone up to 12 inches belt. In the *Kashima* and *Katori* the belt at the ends is not reduced below 6½ inches. In the *Satsuma* it is 6 inches forward and 4 inches aft, the reduced thickness being compensated for by increased area. Thus, the *Kashima* forward has a belt of 7½ feet wide, surmounted by a 7½-foot strake of thinner armor, whereas the *Satsuma* has a 15-foot belt of 6-inch armor. This is in accordance with, and was somewhat in advance of, modern practice—the tendency nowa-days being to reduce maximum thickness in favor of increased area.

At a casual glance the *Satsuma* might appear to be a *Lord Nelson.* Actually, however, she differs materially from that ship in having a main-deck battery of 6-inch guns. As originally designed, this main-deck battery was to have consisted of 4.7's, but at a somewhat late stage 6-inch guns were substituted.

Eight are carried in the center battery. Two more guns are carried right forward on main deck and two others right aft. There is some doubt as to whether the substitution of 6-inch included these guns, and, according to some reports, only 12-

Name.	*Satsuma.*	*Lord Nelson* class.	*Danton* class.	*Radetsky* class.	*Dreadnought.*	*Nassau* class.	*Minas Geraes* class.	*Cuniberti* ideal ship (1903).
Displacement, tons.....	19,250	16,500	18,400	14,500	17,900	18,500	19,281	19,000
Length, feet.....	479	435	475½	456	520	470	530	500
Beam, feet.....	83½	79½	84½	82	82	89	83	...
Max. draught, feet.....	28½	27 (mean).	27½	26½ (mean).	31	26½	25 (mean).	...
Laid down.....	May, 1905.	Nov., 1904.	Feb., 1908.	Dec., 1907.	Dec., 1905.	July, 1906.	1907.	...
Launched.....	Nov., 1906.	1906.	Jan., 1909.	July, 1909.	Feb., 1906.	Mar., 1908.	Sept., 1908.	...
Completed.	1910.	1908.	1912 (to be)	1910 (to be)	Oct., 1906.	Oct., 1909.	Jan., 1910.	...
Armament.....	4 12-inch. 12 10-inch. 12 6-inch. 4 12-pdr.	4 12-inch. 10 9.2-inch. 15 12-pdr. 16 3-pdr. 2 Maxims.	4 12-inch. 12 9.4-inch. 16 12-pdr. 10 3-pdr.	4 12-inch. 8 9.4-inch. 20 4.1-inch.	10 12-inch. 27 12-pdr. ...	12 11-inch. 12 6-inch. 16 24-pdr.	12 12-inch, 22 4.7-inch. 8 3-pdr.	12 12-inch. 12 12-pdr.
Torpedo tubes.....	5	5	2	3	5	6	0	4
Horsepower.....	18,500	16,750	22,500	20,000	23,000	20,000	23,500	...
Speed, knots.....	20	18.5	19.4	20	21	19	21	24
Machinery	Triple-expansion.	Triple-expansion.	Parsons turbine.	Triple-expansion.	Parsons turbine.	Triple-expansion.	...	...
Boilers.....	Miyabara.	Babcock & Yarrow.	Belleville & Niclausse.	Yarrow.	Babcock.	Schulz.	Babcock.	...
Armor belt, max., ins..	9	12	10	9	11	10	9	12

pounders are mounted in these positions. Four other 12-pounders are carried amidships on the superstructure around the funnels. It is worthy of note that the Japanese have refrained from mounting any guns whatever on the superstructures around the fore and aft conning towers, as is the general practice in all other navies. It is understood that in the war on one or two occasions the presence of guns' crews in the neighborhood of conning towers was found inconvenient. It has also been stated that guns firing in close proximity to the men operating searchlights was found undesirable.

A feature of the *Satsuma* is the fact that her 12-inch turrets are something like half as large again as the turrets for the 10-inch guns. These main turrets have also a somewhat higher command.

We append a table, showing how the *Satsuma* compares with a variety of other ships.—"The Engineer."

OBITUARY.

CHIEF ENGINEER GARDINER C. SIMS.*

The many friends of the genial Gardiner C. Sims, formerly a Chief Engineer in the Navy, will regret to learn that he died at his home in Providence on the 19th of March.

He died of Bright's disease, after a long and painful illness. At the time of his death he was President of the Wm. A. Harris Steam Engine Works, at Providence, which position he had held for a number of years. He was the inventor and patentee of the famous Armington and Sims high-speed engine, the engine first used to drive the Edison dynamo electric machines, and which had so much to do with making the dynamos a success. That engine has not, even at this day, been surpassed for excellence, but cheaper engines have taken its place.

Mr. Sims was born in the village of Niagara Falls in 1845, where he received his early education. He served an apprenticeship in the locomotive works of the Hudson River Railroad, and finally became their chief draughtsman and designer. He was at once an engine builder and designer.

In 1898 he left the Armington and Sims Company to enter the Navy, being appointed a chief engineer, and was assigned to duty on board the repair ship *Vulcan*. The ship was equipped with a pattern shop, foundry, smithy, copper shop and machine shop. For this place Mr. Sims was especially well equipped.

Upon leaving the naval service he received a very complimentary letter from the Secretary of the Navy, in recognition of his services.

He then returned to Providence, where he first became

* Written by Rear Admiral Geo. W. Baird, U. S. N.

general manager and afterwards president of the William A. Harris Steam Engine Works, which position he held up to the time of his death.

Mr. Sims was a member of the Society of Naval Architects and Marine Engineers, of the Mechanical Engineers, the American Society of Naval Engineers, of the Rhode Island Historic Society, the Squantum Club and the Free Masons.

Socially he was one of the most genial of men; though fond of club life, he was abstemious to a fault; he never tasted a glass of liquor in his life. Mr. Sims leaves a widow and one son; the latter is now a student at Brown University.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

SPRING MEETING, ATLANTIC CITY, N. J., MAY 31-JUNE 3, 1910.

The Spring Meeting, which will be held in Atlantic City, N. J., May 31–June 3, 1910, is the regular semi-annual meeting of the Society.

The joint meeting in England in July is an additional meeting.

HEADQUARTERS.

The headquarters of the meeting will be the Marlboro-Blenheim, situated at the central point of the board walk. The Registration Office will be located in the hotel, on the board-walk floor, and the professional sessions will be held in the solarium on the office floor.

PROGRAM.

Tuesday Afternoon and Evening, May 31.—Informal reunion of members in the parlors of the Marlborough-Blenheim.

Wednesday, June 1, 10 A. M.—Professional Session.—Business meeting; Reports of Committees and of Tellers of Election; new business.

Papers on Machine Construction and Operation: The Shockless Jarring Machine, Wilfred Lewis; A Comparison of Lathe Headstock Characteristics, Prof. Walter Rautenstrauch; The Strength of Punch and Riveter Frames made of Cast Iron, Prof. A. L. Jenkins; Improved Methods in Finishing Stay Bolts and Straight and Taper Bolts as Used in Locomotives, C. K. Lassiter.

Wednesday Afternoon and Evening.—This afternoon is left unassigned to give opportunity for sightseeing. Roller

chairs for the board walk will be available for the visiting members and guests, through the courtesy of the Local Committee.

In the evening entertainment on the Steel Pier has been provided by the Committee.

Thursday, June 2, 10 A. M.—Gas Power Section.—Business meeting and reports of committees.

Papers: A Regenerator Cycle for Gas Engines using Sub-Adiabatic Expansion, Prof. A. J. Frith; Gas Engines for Driving Alternating-Current Generators, H. G. Reist; Two Proposed Units of Power, Prof. Wm. T. Magruder; Some Operating Experiences with a Blast Furnace Gas Power Plant, H. J. Freyn.

Thursday, 2 P. M.—Professional Session. Miscellaneous papers: The Mechanical Engineer and the Textile Industry, H. L. Gantt; The Elastic Limit of Manganese and other Bronzes, J. A. Capp; The Hydrostatic Chord, R. D. Johnson; The Resistance of Freight Trains, Prof. Edw. C. Schmidt.

Thursday, 9 P. M.—Reception, followed by conferring of Honorary Membership on Rear Admiral George W. Melville, U. S. N., Retired. A brief address will be made by Admiral Melville. Dancing and refreshments.

Friday, June 3, 10 A. M.—Professional Session. Papers on Power Transmission: Improvements in Lineshaft Hangers and Bearings, Henry Hess; Experimental Analysis of a Friction-clutch Coupling, Prof. Wm. T. Magruder; An Improved Absorption Dynamometer, Prof. C. M. Garland; Critical Speed Calculation, S. H. Weaver.

BOOKS REVIEWED.

COURS DE MACHINES MARINES, DEUXIÈME PARTIE. MACHINES ALTERNATIVES, TURBINES MARINES, MOTEURS A COMBUSTION INTERNE. PAR L. JAUCH ET A. MASMÉJEAN, OFFICIERS MÉCANICIENS, PROFESSEURS A L'ÉCOLE DES MÉCANICIENS DE TOULON, ANCIENS ÉLÈVES DES ÉCOLES NATIONALES D'ARTS-ET-MÉTIERS. (In French.) J. ALTÉ, Toulon. Price, 30fr., with Plates.

This volume constitutes the second part of the course in marine engines at the Toulon Engineering School. The volume on boilers and their accessories, by the same authors, reviewed in the May, 1909, number of the JOURNAL, constitutes the first part of the same work.

This is a comprehensive work of 869 pages, with 545 figures in the text and an accompanying volume of 51 large plates. The illustrations are of particular excellence, consisting of half tones, or shaded drawings, bringing out the details with great clearness.

The work opens with an introductory of 117 pages on the materials of engineering, explaining briefly but clearly the processes of manufacture, the special methods necessary in working different metals, and the methods employed in testing steel, iron and the alloys.

The reciprocating engine is treated in 471 pages of the work, including the theory of the steam engine, theoretical and practical design, with many illustrations giving actual recent practice in several well known vessels.

The chapters on the marine steam turbine comprise 148 pages, in which the design of turbine blading is treated mathematically for both the reaction and impulse types. The arrangement of machinery and design of screw propellers

for turbine-driven vessels are also treated. The different types of torsionmeters in use are explained briefly, but understandingly. Details of construction of the de Laval, Rateau, Curtis, Parsons and Zoelly turbines are given, while the Parsons marine installation is treated at considerable length.

The last 133 pages are on internal-combustion engines. The theory and design of engines of this type are given briefly, followed by discussion of the different fuels employed and descriptions of the various detail parts of such engines. Detailed instructions are given for the care and management of the types of internal-combustion engines found in service afloat. The work closes with discussions of the possibilities of explosive engines as motive power for war vessels and the progress made in the design of gas turbines.

This work will prove a valuable addition to the library of an engineer who is familiar with the French language.

ASSOCIATION NOTES.

AWARD OF PRIZE ESSAY.

At a meeting of the Society held on March 4, 1910, the votes for prize essay for the year 1909 were opened and counted.

The article, "Screw Propeller Criticism and Notes on Screw Propeller Design Based on Actual Standardization Trial Results of U. S. Vessels," by Commander C. W. Dyson, U. S. N., received a majority of the votes received, and the prize for the year 1909 was accordingly awarded to Commander C. W. Dyson.

ANNOUNCEMENT.

Instead of awarding an annual prize essay, the Council, at a meeting held on May 14, decided that it would be more advantageous to pay liberally for contributions of particular merit and to award a prize to such essays as may be deemed especially meritorious, whenever such may be contributed.

All Members and Associates are especially invited to submit original contributions on engineering matters of interest.

It is hoped that greater interest in the matter of sending contributions for publication in the JOURNAL may be shown by the members of the Society, and it is manifest that the larger and better the supply of material received the more valuable will the pages of the JOURNAL become.

SOCIETY BANQUET.

The Society banquet, held at Rauscher's, Washington, D. C., on May 7th, was a most successful event. About 150 were present.

The President of the Society, Engineer-in-Chief H. I. Cone,

U. S. N., after welcoming the guests and delivering a few well-chosen remarks, dwelling on the essential features of naval engineering, introduced the toastmaster, Mr. Walter M. McFarland, who then took charge and introduced the various speakers with many well-chosen remarks.

The Hon. Secretary of the Navy responded to the toast, "The Navy and Engineering." His remarks, which called attention to the various engineering developments now in progress and to the importance that engineering bears in efficiency of the Navy, were received with great enthusiasm.

The Hon. L. P. Padget responded to the toast, "What the People Demand of the Navy." Rear Admiral Richard Wainwright responded to "The Fleet." Hon. E. W. Roberts to "The Navy and Congress," and Hon. Joseph G. Cannon responded to the toast, "The Navy and the People."

Besides these regular speakers, short addresses were given by Rear Admiral Swift, "Mr. Geo. Westinghouse," Editor C. W. Baker, of "Engineering News," and Surgeon General Chas. F. Stokes.

The guests also had the pleasure of listening to the entertaining recitations of Mr. Jack Armour and the singing of Mr. Wm. Murray and Mr. Bert Self, several of the songs being especially dedicated to the engineering profession.

The banquet, on the whole, was a most successful affair, and the large attendance and the many distinguished engineers and public men who honored the Society by their presence are matters of much gratification.

Following is a list of the members and guests who attended the banquet:

Comdr. W. H. Alderdice, U. S. Navy; Mr. H. L. Aldrich, Brig. Gen. J. B. Aleshire, U. S. Army; Mr. Ernest H. B. Anderson, Comdr. Philip Andrews, U. S. Navy; Mr. Jack Armour and Mr. D. R. M. de Azevedo.

Mr. C. W. Baker, Mr. R. O. Bailey, Lieut. H. A. Baldridge, U. S. Navy; Hon. Arthur Bates, M. C.; Capt. F. W. Bartlett, U. S. Navy; Mr. C. A. Blomberg, Mr. Scott C. Bone, Mr. Allen R. Boyd, Mr. George K. Bradfield, Mr. Joseph

A. Breckons, Mr. L. W. Brigham, Hon. Thomas Butler, M. C.; Lieut. Comdr. H. V. Butler, U. S. Navy.

Mr. F. E. Caldwell, Hon. Joseph Cannon, Speaker of the House; Chief Constructor W. L. Capps, U. S. Navy; Mr. R. E. Carney, U. S. Navy; Comdr. C. A. Carr, U. S. Navy; Mr. R. H. Chappell, Lieut. A. T. Church, U. S. Navy; Eng.-in-Chief H. I. Cone, U. S. Navy; Civil Eng. F. H. Cooke, U. S. Navy; Mr. Frank C. Corbett, Mr. H. E. Cornell, Mr. C. M. Coryell, Lieut. Ormond L. Cox, U. S. Navy; Mr. Alleston F. Cushman.

Lieut. R. C. Davis, U. S. Navy; Hon. A. F. Dawson, Mr. Charles E. Delanoy, Lieut. W. G. Diman, U. S. Navy; Lieut. H. C. Dinger, U. S. Navy; Lieut. J. C. Dorry, U. S. R. C. S.; Lieut. S. H. R. Doyle, U. S. Navy; Mr. Edwin Duff, Comdr. C. W. Dyson, U. S. Navy.

Lieut. P. B. Eaton, U. S. R. C. S.; Maj. Gen. George Elliott, U. S. M. C.; Mr. W. L. R. Emmet, Mr. R. L. Ettenger, Hon. W. F. Englebright, M. C.

Capt. Frank F. Fletcher, U. S. Navy; Rear Admiral John D. Ford, U. S. Navy; Comdr. A. B. Fry, N. Y. N. M.

Mr. W. B. Gifford, Mr. C. P. Glein, Prof. Frederick A. Goetze, Mr. E. W. Graef, Mr. H. B. Gregory, Mr. Bernard R. Green, Capt. R. S. Griffin, U. S. Navy.

Lieut. Comdr. John Halligan, Jr., U. S. Navy; Mr. Charles F. Heitzman, Lieut. Robert Henderson, U. S. Navy; Mr. W. E. Hicks, Hon. Charles D. Hilles, Lieut. Comdr. U. T. Holmes, U. S. Navy; Rear Admiral R. C. Hollyday, U. S. Navy; Prof. Alex. C. Humphreys, Rear Admiral R. Inch, U. S. Navy.

Mr. E. N. Jansen and Mr. J. E. Jenks.

Mr. L. Katzenstein, Mr. M. C. Katzenstein, Mr. William Katzenstein, Mr. J. W. Kellog, Mr. William R. King, U. S. Navy; Capt. T. W. Kinkaid, U. S. Navy; Lieut. H. Kotzschmar, U. S. R. C. S.; Mr. H. J. Kunzig.

Mr. Inglis Leslie, Hon. H. C. Loudenslager, M. C.; Lieut. Comdr. Frank Lyon, U. S. Navy.

Hon. J. R. Mann, M. C.; Hon. George von L. Meyer, Sec-

retary of the Navy; Lieut. S. I. M. Major, U. S. Navy; Mr. Frank Martin, Mr. A. M. P. Maschmeyer, Mr. H. M. McMillan, Engineer-in-Chief C. A. McAllister, U. S. R. C. S.; Mr. Walter M. McFarland, Mr. A. M. LeP. Mesney, Mr. J. F. Metten, Mr. H. T. Morningstar, Mr. William Murray.

Lieut. Comdr. C. N. Offley, U. S. Navy, and Hon. J. Van Vechten Olcott, M. C.

Hon. L. P. Padgett, M. C.; Mr. C. H. Paine, Mr. Elmer E. Paine, Lieut. Comdr. L. C. Palmer, U. S. Navy; Mr. N. B. Payne, Mr. C. H. Peabody, Paymaster C. J. Peoples, U. S. Navy; Rear Admiral W. P. Potter, U. S. Navy; Hon. Pitman Pulsifer.

Mr. A. P. Rankin, Lieut. Comdr. M. E. Reed, U. S. Navy; Commodore I. S. K. Reeves, U. S. Navy; Lieut. J. O. Richardson, U. S. Navy; Mr. F. H. Rittenour, Lieut. S. M. Robinson, U. S. Navy; Lieut. Comdr. J. K. Robison, U. S. Navy; Hon. Ernest W. Roberts, M. C.; Mr. Stephen Rozycki, Capt. W. G. Ross, U. S. R. C. S.; Mr. R. R. Row, Mr. Philip J. Ryan.

Mr. F. X. Sadlier, Mr. Bert Self, Lieut. Comdr. W. R. Sexton, U. S. Navy; Lieut. F. L. Sheffield, U. S. Navy; Mr. George C. Shepard, Mr. W. F. Sicard, Hon. G. Bascom Slemp, M. C.; Mr. W. Smith, Comdr. W. Strother Smith, U. S. Navy; Pay Director J. N. Speel, U. S. Navy; Mr. J. B. Spencer, Dr. S. W. Stratton, Mr. R. B. Strasburger, Mr. John A. Stewart, Mr. Alex. Stewart, Surgeon General Charles F. Stokes, U. S. Navy; Rear Admiral William Swift, U. S. Navy.

Lieut. W. B. Tardy, U. S. Navy; Mr. J. Knox Taylor, Lieut. E. S. Theall, U. S. M. C.; Comdr. Emil Theiss, U. S. Navy; Lieut. H. V. Tomb.

Rear Admiral Richard Wainwright, U. S. Navy; Mr. Ernest G. Walker, Mr. J. C. Wallace, Mr. George E. Walton; Capt. John Q. Walton, U. S. R. C. S.; Mr. Charles Ward, Lieut. W. B. Wells, U. S. Navy; Mr. George Westinghouse, Hon. William H. Wiley, M. C.; Lieut. Comdr. W. R. White,

U. S. Navy; Comdr. W. W. White, U. S. Navy; Mr. S. B. Whiting.

Mr. William F. Yates.

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